WORTHINGTON PUMP HANDBOOK

DESIGN, CONSTRUCTION AND APPLICATION OF ALL COMMERCIAL TYPES OF PUMPS

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WORTHINGTON ENGINEERS

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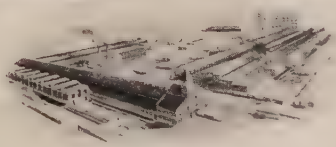

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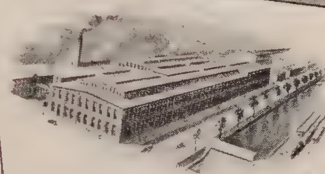

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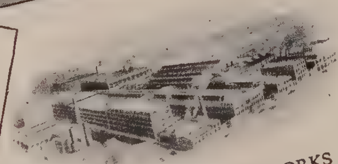

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FOREWORD

In 1840 Henry R. Worthington placed his first pump on the market and thereby launched a great industry. This pump which he had invented was, moreover, the first direct-acting pump ever built, It was entirely different from anything previously used and was so perfectly simple and efficient that it revolutionized the whole course of pumping practice. Type after type of these pumps has since been devised to cover the needs of a constantly widening field, with the result that the name Worthington is recognized the world over as standard in pump development.

Thus it was that Worthington came to play such an important part, not only in the history of pump development, but also in rendering possible many other engineering advances by co-operation with producers of other types of mechanical equipment requiring pumping service. It is an interesting fact that industrial progress and the general welfare of the human race have been largely dependent on advances in the science of pumping, and since 1840 Worthington has led in this science. Year after year the fame of Worthington pumps has spread, until today they are known and used all over the civilized world.

This book is a text on Worthington pumps. It is more; it is a useful text on pumping. It has been written by Worthington engineers, each contributing material pertaining to that branch of pump engineering which is his daily work, and is offered as an authoritative and practical book, worthy a place in the reference library of any engineer, student, or user of mechanical equipment. In a book of this size, and containing so much tabular matter and so many mathematical formulae, it is almost inevitable that errors will creep into the first edition. Notification will be appreciated of any errors that may be discovered, so that they may be corrected in subsequent editions, and suggestions as to material to be included or expanded in the next edition will be welcomed. Lest this book create the impression that Worthington makes nothing but pumps, it should be stated that the Corporation also manufactures air, ammonia and gas compressors; Diesel oil engines and

gas engines; surface, barometric and jet steam condensers; steam air-ejectors and other vacuum-producing machinery; water and oil meters; stationary and locomotive feedwater heaters. The Worthington line of steam power-plant auxiliaries is the most complete in existence. Literature on any of these products will be gladly forwarded on request.

The Worthington engineering staff is at the service of industry. Inquiries are solicited on problems connected with pumping, or any of the other services performed by Worthington equipment.

WORTHINGTON PUMP AND MACHINERY CORPORATION

CONTENTS

SECTION I

General Theory and Units

(Figures refer to paragraph numbers)

SECTION I

General Theory and Units

1. Pumps.—A pump may be broadly defined as a machine for the transfer of liquids or gases from one location to another through pipes. To accomplish this result the principles of suction and forcing may be used; in practice, generally a combination of the two.

2. Liquids and Gases.—A liquid is an inelastic fluid, keeping practically the same volume, but taking always the form of the containing vessel. A gas is a highly elastic fluid and tends to expand to an indefinitely large volume. As this book will treat solely with the problems of pumping liquids, we will refrain from discussing the characteristics of all gases excepting steam. Liquids vary greatly in their character. They may be thin and of low viscosity, like water, gasoline and alcohol, or they may be thick and of high viscosity, like tar, molasses and certain kinds of crude oils.

3. Viscosity of Liquids.—Viscosity is the resistance of a liquid flowing under stress and is due to molecular fluid friction. It is that property of liquids which makes it hard to stir cold molasses, and which brings the coffee in a cup to rest after it has been stirred. Viscosity varies through wide limits, and in general decreases as the temperature of the liquid increases, which feature is of great advantage when pumping tar, asphalt base crude oils, etc. This is fully explained in Section 7, par. 15. to par. 27.

4. Basic Units of Liquid Measure.—The U. S. gallon and the cubic foot are the basic units used by hydraulic engineers for the measurement of liquid quantities. **The U. S. gallon** contains 231 cu. in. and is equivalent to 0.134 cu. ft., or 1 cu. ft. = 7.481 gal. The U. S. gallon is the standard used throughout this work.

5. Inquiries received from foreign countries sometimes refer to the **British Imperial gallon,** which contains 277.27 cu. in. To reduce British to U. S. gallons multiply by 1.2. To convert U. S. into British imperial gallons divide by 1.2.

6. Many **commercial standards** of liquid measure have been set up by different industries, all having the gallon or the cubic foot as a basis. Throughout the oil industry the barrel (42 gal.) is the

most used standard. In irrigation the "acre-foot" or "acre-inch" and in some States the "miner's-inch" are the standards. The "second-foot" is another standard extensively used.

7. The **acre-foot** is the amount of water which will cover one acre of land to a depth of one foot and is equal to 43,560 cu. ft. = 325,-872.3 gal. An **"acre-inch"** is equal to one-twelfth of an "acre-foot" = 3630 cu. ft. or 27,154.2 gal.

8. The **miner's inch** varies in different localities and is not very satisfactory as a unit of measure. For example, in parts of California and in Arizona 40 miner's inches = 1 cu. ft. per sec. or 1 miner's inch = 12 gal. per min. In New Mexico, Oregon and Washington 50 miner's inches = 1 cu. ft. per sec. or 1 miner's inch = 9 gal. per min. In Colorado 38.4 miner's inches = 1 cu. ft. per sec. or 1 miner's inch = 11.75 gal. per min. The miner's inch is governed by the velocity of the flow, and as this is determined by the head above the aperture, or the grade of the flume, it is advisable to use this form of measurement with discretion.

9. Another unit of measure extensively used in hydraulic work is the **"second-foot,"** which is the delivery of one cubic foot of water per second. This form of measurement is controlled by a flume one foot wide, one foot deep and so inclined as to give a velocity to the water of one foot per second, and is equal to a capacity of 448.8 gal. per min.

10. Effects of Liquids on Metals.—Many liquids have a solvent action or a corrosive action on certain metals. For instance, naphtha has a solvent action on cast iron and tends to soften the metal, thereby causing a weakness in pump castings. Mine waters which contain acids have a corrosive action on cast iron and on bronze containing zinc. It is, therefore, of the utmost importance that the metal used in the liquid end of a pump be given careful consideration. In many cases it is necessary to use a liquid end composed entirely of iron or steel; in other cases a combination of iron and bronze; in others all parts of the liquid end must be of bronze. For example, creosote oil requires all iron and steel construction. Acid water and certain kinds of acids can be handled with a liquid end composed entirely of bronze.

11. Acids will rapidly corrode magnesium, manganese, aluminum and zinc. The use of bronze containing these metals should be

avoided when selecting the proper composition to use in the liquid end of pumps for handling acids or liquids containing acid.

12. Worthington has adopted as a **standard acid-resisting metal** a bronze composed of copper, tin and lead in the proper proportions. The long operating life of pumps having liquid ends made from this acid-resisting bronze has fully demonstrated its value in chemical and industrial processes where acids are to be pumped and in mine service where acidulous water is encountered.

13. However, as nearly all acids will slowly corrode bronze of any composition, the operating life of a pump handling acids, or liquids containing acids, cannot be guaranteed.

14. In some types of pumps for acid work it is possible to make the entire liquid end of a hard lead; in others, cast iron, with a lead or wood lining. The material to be used for pumps handling various liquids is given in the tables under par. 140.

16. Principles of Mechanics and Hydraulics.—In order to select the right size of pump, it is usually necessary to make certain mathematical calculations. To enable the user of this book to make such calculations for himself or to check up the work of another, we will review some of the fundamental principles of mechanics and hydraulics. These subjects cannot be dealt with extensively in a volume of this size but enough can be briefly considered to cover most cases that come up in pumping problems.

17. The majority of terms used in this work are common words in daily use, but they have much narrower and more restricted meanings when used in their technical sense. Definitions of the terms, used together with the units of measurement adopted for each term in the United States and in other English-speaking countries, are:

18. Force is anything that will produce, change or destroy motion, or that tends to do so. The unit of measurement is pounds.

19. Motion is a change of position or location. The unit of measurement is feet or miles.

20. Velocity is the time rate of change of position, or time rate of motion. Velocity is measured in feet per second, feet per minute, or miles per hour.

21. Acceleration is the time rate of change of velocity. Acceleration is expressed in the fundamental units of length and time and is measured in feet per second per second. To illustrate, suppose a body should start from rest and increase its velocity one foot per second per second for a minute. Its velocity at the end of the minute would be 60 ft. per sec. Since it would acquire in one second a velocity of one foot per second, and in one minute a velocity of 60 ft. per sec., its acceleration would be one foot per sec. per sec., or 60 ft. per sec. per min.

22. The **mass** of a body is the measure of its inertia; that is, of the resistance which a body offers to motion or change of motion. The value of mass is always represented in formulas as

$$m = \frac{w}{g} \text{ or } \frac{w}{32.16}$$

where m = mass; w = weight and g = acceleration due to gravity = 32.16.

23. Momentum is inertia of motion; sometimes called quantity of motion, and is the product of the mass and the linear velocity of the moving body or

$$M = m\,v$$

where M = momentum; m = mass and v = velocity.

24. Inertia is the property of matter by virtue of which it persists in its state of either rest or of uniform motion and resists any attempt to change that state. Inertia in the case of a stationary body has no unit of measurement. The inertia of a moving body is its momentum and is expressed by the formula

$$M = \frac{w}{g}v.$$

25. Work is the overcoming of a resistance through a distance. Work is measured in foot-pounds.

26. A **foot-pound** is the work done by a pound of force working through a distance of one foot.

27. Power is the time rate of doing work. The unit of power commonly used is the **horsepower** (abbreviation hp.); it is the rate of doing work equal to 33,000 ft-lb. per min., or 550 ft-lb. per sec. In electrical work the unit of power commonly used is the **watt.** It is work done at the rate of one **joule** per second. A **kilowatt** is 1000 watts. One horsepower is equivalent to 746 watts.

One kilowatt is about 1.34 hp. For approximate calculations, to convert kilowatt to horsepower add one-third; to convert horsepower to kilowatt subtract one-fourth. Example, 90 kw. plus 1/3 = 120 hp., and 200 hp. minus 1/4 = 150 kw.

28. Energy is the capacity for doing work. The unit of measurement is foot-pound. Energy is of two kinds, potential and kinetic.

29. Potential energy may be defined as the energy of position or simply as stored energy. Energy may be stored in mechanical displacements or in chemical and physical changes in a body. A coiled clock spring, a storage battery or a reservoir of water on a hilltop are examples of potential or stored energy.

30. Kinetic energy is the energy of a body in motion. The motion may be molecular and invisible, as the heat energy in a piece of hot metal. A revolving flywheel of an engine, and the water discharged from a hose nozzle, represent kinetic energy.

31. Kinetic energy must not be confused with force. A moving body carries with it a definite quantity of energy, but it exerts no force until it encounters resistance. Energy is then transferred to the resisting body; force is exerted only during this transfer. The kinetic energy of a moving body is equal to one half its mass multiplied by the square of its velocity in feet per second. Expressed as a formula

$$\textit{Kinetic energy} = \tfrac{1}{2}\, mv^2 = \frac{wv^2}{2\,g} = \frac{wv^2}{64.32}$$

where m = mass; v = velocity in ft. per sec.; w = weight and $g = 32.16$.

32. Definitions of Thermal Quantities.—The **British Thermal Unit** (B.t.u.) is the heat required to raise the temperature of one pound of water one degree Fahrenheit (average between 32 and 212 deg.).

33. The **large calorie** is the amount of heat required to raise the temperature of one kilogram of water one degree centigrade. It is 1000 times as large as the small calorie. As one kilogram is

equal to 2.2046 lb. and one degree centigrade is equal to 9/5 degrees Fahr. the large calorie is $2.2046 \times 9/5 = 3.9683$ times as great as the British thermal unit.

34. The **small calorie** is the amount of heat required to raise the temperature of one gram of water one degree centigrade. A small calorie is 1/1000 of a large calorie and is 0.003965 times the British thermal unit.

35. The **heat of combustion** of a substance is the number of British thermal units of heat evolved during the combustion of one pound of the substance. Using the metric system, the heat of combustion of a substance is the number of large or small calories of heat evolved during the combustion of one kilogram or one gram of the substance. Large calories per kilogram $\times 1.8 =$ B.t.u. per pound.

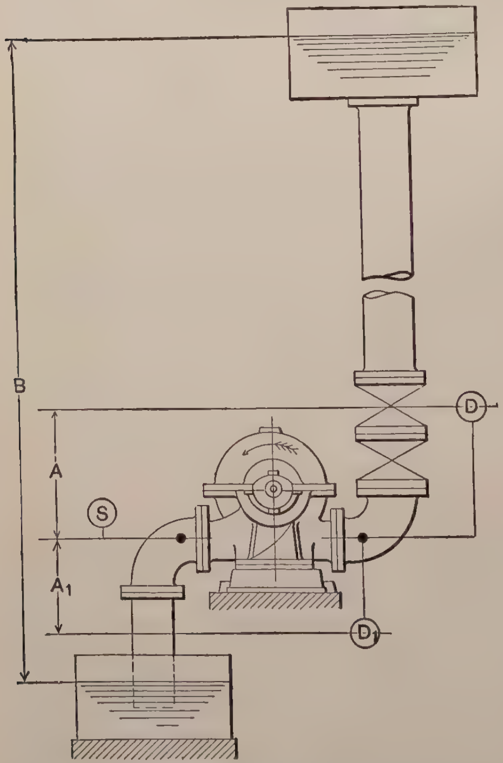

FIG. 1. Location of gages for measuring total dynamic head with suction below pump intake. Centrifugal Pumps.

36. The heat of formation of a substance is the number of calories of heat evolved or absorbed when a gram molecular weight of the substance is formed. When heat is absorbed the value found is negative.

37. The **melting point** of the substance is the temperature at which the solid and liquid forms are capable of existing together in equilibrium.

38. The **boiling point** of a liquid is the highest temperature at which the liquid and

its pure vapor can exist together in equilibrium. As this temperature varies with the pressure, the boiling point of a liquid may be defined as the temperature at which it gives off vapor at a pressure equal to that sustained by the surface of the liquid. As the pressure on a liquid is increased or decreased the boiling point is increased or decreased.

39. The **specific heat** of a substance is the ratio of the number of thermal units necessary to raise the temperature of the substance one degree, divided by the number of thermal units necessary to raise the same weight of water at 60 deg. F. one degree. In the metric system it is the number of thermal units required to raise the temperature of one gram of a substance one degree centigrade.

40. The **heat of fusion** of a substance is the number of thermal units required to change a unit mass of the solid at its melting point into liquid at the same temperature.

41. The **heat of vaporization** of a liquid is the number of heat units required to change a unit mass of the liquid at its boiling point into vapor at the same temperature.

42. Capacity of Pumps.—The first thing to be considered in selecting a pump is, of course, the capacity required in gallons per minute. Rules for calculating the capacity of the different types of positive displacement and centrifugal pumps are given in chapters describing each type.

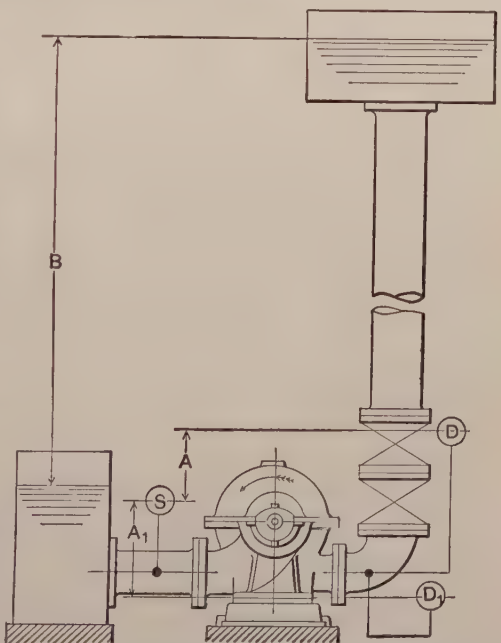

FIG. 2. Location of gages for measuring total dynamic head with suction above pump intake. Centrifugal Pumps.

43. The next consideration is the **total dynamic head** (t.d.h.) against which the pump is to work. The term "head" is here used to designate the total vertical distance or its equivalent in feet or pounds per square inch through which the liquid is transferred by the pump. Fig. 1 illustrates a centrifugal pump with suction lift and Fig. 2 a centrifugal pump with a positive suction head.

44. To Determine Total Dynamic Head.—When using gages for determining the t.d.h. against which the pump is operating, the gages should be connected as close to the suction and discharge flanges as practical. The discharge pressure gages (D) and (D_1) (see Fig. 1 and Fig. 2) should be read as if reading at the center of the gage. The vacuum gage (S) should be read as if reading at the point where it connects to the pipe. The t.d.h. on the pump is the algebraic sum of the two gage readings, the difference in altitude between them and the difference in the velocity head between the suction and discharge pipes. This can be expressed as a formula where

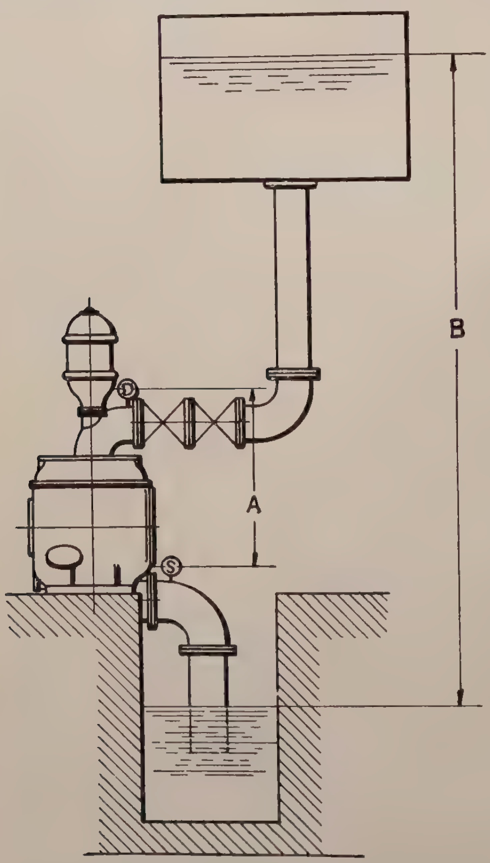

FIG. 3. Location of gages for measuring total dynamic head with suction below pump intake. Reciprocating Pumps.

t.d.h. = total dynamic head

D = discharge head in feet

S = Suction head or lift in feet
A = distance between gages
V_1 = velocity in discharge pipe in feet per second
V_2 = velocity in suction pipe in feet per second
V_{dh} = velocity difference head in feet = $\left(\frac{V_1^2 - V_2^2}{2g}\right)$
g = acceleration due to gravity = 32.16
B = Static head in feet.

For a centrifugal pump with suction lift (see Fig. 1)

1. $t.d.h. = D + S + A + V_{dh}$
2. $t.d.h. = D_1 + S - A_1 + V_{dh}$
3. $t.d.h. - B$ = Friction head

For a centrifugal pump with a positive suction head (see Fig. 2)

1. $t.d.h. = D - S + A_1 + V_{dh}$
2. $t.d.h. = D_1 - S - A + V_{dh}$
3. $t.d.h. - B$ = Friction head

45. The total dynamic head for a reciprocating pump (see Fig. 3 and Fig. 4) is determined by the same method. The V_{dh} however may be neglected for all practical purposes as the velocities of the liquids are low and the total hydraulic losses in a well-designed direct-acting pump seldom exceed 1.6 ft. Table, par. 46, giving the pressure of water will be found useful in converting the total dynamic head in feet to pressure in pounds per sq. inch, or vice versa.

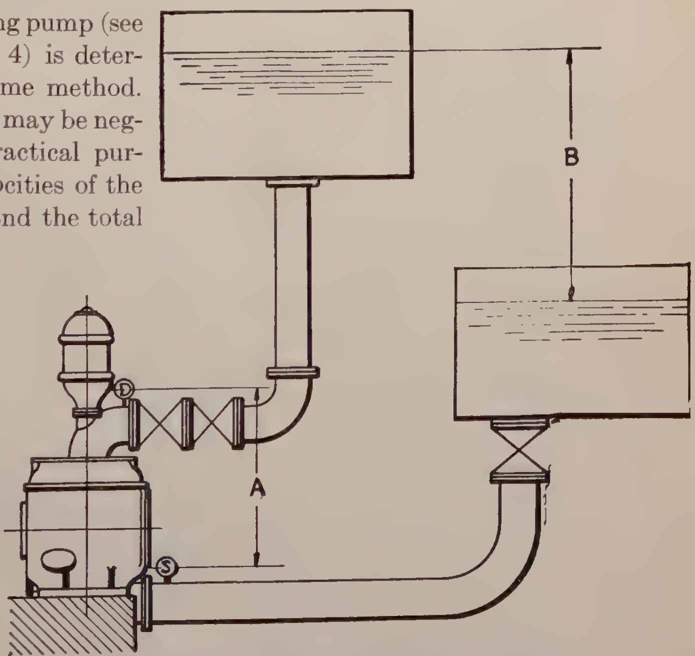

FIG. 4. Location of gages for measuring total dynamic head with suction above pump intake, Reciprocating Pumps.

46. PRESSURE OF WATER

Feet Head	Pressure Per Sq. In.	Feet Head	Pressure Per Sq. In.	Feet Head	Pressure Per Sq. In.	Feet Head	Pressure Per Sq. In.	Feet Head	Pressure Per Sq. In.
1	0.43	33	14.29	65	28.15	97	42.01	129	55.88
2	0.86	34	14.72	66	28.58	98	42.45	130	56.31
3	1.30	35	15.16	67	29.02	99	42.88	131	56.74
4	1.73	36	15.59	68	29.45	100	43.31	132	57.18
5	2.16	37	16.02	69	29.88	101	43.75	133	57.61
6	2.59	38	16.45	70	30.32	102	44.18	134	58.04
7	3.03	39	16.89	71	30.75	103	44.61	135	58.48
8	3.46	40	17.32	72	31.18	104	45.05	136	58.91
9	3.89	41	17.75	73	31.62	105	45.48	137	59.34
10	4.33	42	18.19	74	32.05	106	45.91	138	59.77
11	4.76	43	18.62	75	32.48	107	46.34	139	60.21
12	5.20	44	19.05	76	32.92	108	46.78	140	60.64
13	5.63	45	19.49	77	33.35	109	47.21	141	61.07
14	6.06	46	19.92	78	33.78	110	47.64	142	61.51
15	6.49	47	20.35	79	34.21	111	48.08	143	61.94
16	6.93	48	20.79	80	34.65	112	48.51	144	62.37
17	7.36	49	21.22	81	35.08	113	48.94	145	62.81
18	7.79	50	21.65	82	35.52	114	49.38	146	63.24
19	8.22	51	22.09	83	35.95	115	49.81	147	63.67
20	8.66	52	22.52	84	36.39	116	50.24	148	64.10
21	9.09	53	22.95	85	36.82	117	50.68	149	64.54
22	9.53	54	23.39	86	37.25	118	51.11	150	64.97
23	9.96	55	23.82	87	37.68	119	51.54	151	65.40
24	10.39	56	24.26	88	38.12	120	51.98	152	65.84
25	10.82	57	24.69	89	38.55	121	52.41	153	66.27
26	11.26	58	25.12	90	38.98	122	52.84	154	66.70
27	11.69	59	25.55	91	39.42	123	53.28	155	67.14
28	12.12	60	25.99	92	39.85	124	53.71	156	67.57
29	12.55	61	26.42	93	40.28	125	54.15	157	68.00
30	12.99	62	26.85	94	40.72	126	54.58	158	68.43
31	13.42	63	27.29	95	41.15	127	55.01	159	68.87
32	13.86	64	27.72	96	41.58	128	55.44	160	69.31

46. PRESSURE OF WATER—Concluded

Feet Head	Pressure Per Sq. In.	Feet Head	Pressure Per Sq. In.	Feet Head	Pressure Per Sq. In.	Feet Head	Pressure Per Sq. In.	Feet Head	Pressure Per Sq. In.
161	69.74	192	83.17	223	96.60	254	110.03	380	164.61
162	70.17	193	83.60	224	97.03	255	110.46	385	166.78
163	70.60	194	84.03	225	97.45	256	110.89	390	168.94
164	71.04	195	84.47	226	97.90	257	111.32	395	171.11
165	71.47	196	84.90	227	98.33	258	111.76	400	173.27
166	71.91	197	85.33	228	98.76	259	112.19	425	184.10
167	72.34	198	85.76	229	99.20	260	112.62	450	195.00
168	72.77	199	86.20	230	99.63	261	113.06	475	205.77
169	73.20	200	86.63	231	100.00	262	113.49	500	216.58
170	73.64	201	87.07	232	100.49	270	116.96	525	227.42
171	74.07	202	87.50	233	100.93	275	119.12	550	238.25
172	74.50	203	87.93	234	101.36	280	121.29	575	249.09
173	74.94	204	88.36	235	101.79	285	123.45	600	259.90
174	75.37	205	88.80	236	102.23	290	125.62	625	270.73
175	75.80	206	89.21	237	102.66	295	127.78	650	281.56
176	76.23	207	89.66	238	103.09	300	129.95	675	292.40
177	76.67	208	90.10	239	103.53	305	132.12	700	303.22
178	77.10	209	90.53	240	103.96	310	134.28	725	314.00
179	77.53	210	90.96	241	104.39	315	136.46	750	324.88
180	77.97	211	91.39	242	104.83	320	138.62	775	335.72
181	78.40	212	91.83	243	105.26	325	140.79	800	346.54
182	78.84	213	92.20	244	105.69	330	142.95	825	357.37
183	79.27	214	92.69	245	106.13	335	145.12	850	368.20
184	79.70	215	93.13	246	106.56	340	147.28	875	379.03
185	80.14	216	93.56	247	106.99	345	149.45	900	389.86
186	80.57	217	93.99	248	107.43	350	151.61	925	400.70
187	81.00	218	94.43	249	107.86	355	153.78	950	411.54
188	81.43	219	94.86	250	108.29	360	155.94	975	422.35
189	81.87	220	95.30	251	108.73	365	158.10	1000	433.18
								1500	649.70
190	82.30	221	95.73	252	109.16	370	160.27	2000	866.30
191	82.73	222	96.16	253	109.59	375	162.45	3000	1299.50

47. VELOCITY HEADS

Calculated from the formula $h = \frac{v^2}{2g}$.

v Feet Per Second	h Feet	v Feet Per Second	h Feet	v Feet Per Second	h Feet	v Feet Per Second	h Feet
2.0	0.06	4.0	0.25	6.0	0.56	8.0	0.99
2.2	0.08	4.2	0.28	6.2	0.60	8.2	1.04
2.4	0.09	4.4	0.30	6.4	0.64	8.4	1.10
2.6	0.10	4.6	0.33	6.6	0.68	8.6	1.15
2.8	0.12	4.8	0.36	6.8	0.72	8.8	1.20
3.0	0.14	5.0	0.39	7.0	0.76	9.0	1.26
3.2	0.16	5.2	0.42	7.2	0.80	9.2	1.31
3.4	0.18	5.4	0.45	7.4	0.85	9.4	1.37
3.6	0.20	5.6	0.49	7.6	0.90	9.6	1.43
3.8	0.22	5.8	0.52	7.8	0.94	9.8	1.49

48. Efficiency of Pumps.—The word efficiency used in connection with pumping machinery has become very ambiguous and has led to many unnecessary disputes in the interpretation of guarantees and contracts. This has been brought about by the fact that the word has been used without modification to designate the efficiency of the pump only, the pump and prime mover, and of the entire plant back to the boiler. This uncertainty can be and should be eliminated by clearly stating that the guaranteed efficiency is to be the mechanical, hydraulic, volumetric, thermal, or overall efficiency of the pump. When it is considered that each of these efficiencies has an entirely different value and that they are independent of each other, the necessity for a clear definition of the term efficiency is apparent.

49. The losses in a pump due to the friction of the moving parts are expressed by the **mechanical efficiency** (μ_m). For a steam-driven pump this is the ratio of the power developed in the liquid end (P_e) to that developed in the steam end (P_s). It is expressed

$$\mu_m = \frac{P_e}{P_s}$$

50. For a pump driven by an electric motor, a steam turbine or other prime mover, the mechanical efficiency (μ_m) is the ratio of the water horsepower (P_w) output to the brake horsepower (P_b) input at the pump coupling or pulley, and is expressed

$$\mu_m = \frac{P_w}{P_b}$$

51. The mechanical efficiency can be determined only by actual test with the different types of pumps. It will be found to vary with the size of the pump, the type and the service.

52. The **hydraulic efficiency** (μ_h) is the ratio of the total head pumped against (h) to the total head pumped against plus all hydraulic losses (h_i). It is expressed

$$\mu_h = \frac{h}{h_i}$$

53. The hydraulic losses comprise all losses, including the velocity head, from the source of supply through the liquid cylinders to the point where the discharge gage is attached.

54. The **volumetric efficiency** (μ_v) is the ratio of the actual discharge in gallons per minute (V) to the plunger or piston displacement (D) expressed in the same unit. It is expressed

$$\mu_v = \frac{V}{D}$$

55. **Thermal efficiency** (μ_t) is the ratio of the heat utilized by the pump in doing useful work to the heat supplied. It is expressed

$$\mu_t = \frac{42.44 \times P \times 60}{S\,(H - h)}$$

Where 42.44 = heat equivalent of one horsepower in B.t.u. per min.
P = horsepower
S = steam consumed in pounds per hr.
H = total heat in one pound of steam at initial pressure
h = Total heat in one pound of feed water

56. Contracts in which the thermal efficiency is guaranteed should be carefully drawn up in respect to the point at which the temperature of the feed water is to be measured. This depends upon whether the feed water is heated by the exhaust of the main engine, the exhaust of auxiliary machinery or by other means. Table, par. 87, gives the properties of saturated steam and

Table, par 88, gives the properties of superheated steam. These will be of value in calculating the thermal efficiencies of pumps.

57. The term **indicated pump efficiency** (μ_{pi}) is not often used. F. F. Nickel in his book, "Direct-Acting Steam Pumps," page 15, gives the indicated pump efficiency as "the ratio of the power developed in raising the water to the power developed in the pump end as disclosed by the indicator diagrams" or

$$\mu_{pi} = \frac{P_w}{P_i}$$

where P_w = water horsepower and P_i = indicated pump horsepower. The value of P_w is calculated from the amount of water (G) actually pumped in gallons per minute and the total head (h). The value of P_i is the horsepower calculated from the displacement (D) and the indicated head (h_i). The indicated pump efficiency can then be expressed

$$\mu_{pi} = \frac{G_h}{D_{hi}}.$$

It is also the product of hydraulic efficiency (μ_h) and the volumetric efficiency (μ_v), or $\mu_{pi} = \mu_h \times \mu_v$ and comprises all the losses in the liquid end.

58. The **total efficiency** (μ) indicates the overall economy of the pump and is expressed

$$\mu = \mu_m \times \mu_h \times \mu_v.$$

59. Horsepower.—The term **horsepower** as used in connection with pumping equipment has more than one application and whenever the term is used it should be clearly stated as pump, water, brake or indicated horsepower, etc.

60. The **pump horsepower** (P_p) or theoretical horsepower as it is sometimes called is equal to the gallons-per-minute displacement (G), times the weight of one gallon of the liquid (w) in pounds, times the total head in feet (h), divided by 33,000; or expressed as a formula

$$P_p = \frac{Gwh}{33000}$$

61. The **water horsepower** (P_w) is based on the amount of water in gallons per minute (G_a) actually discharged by the pump.

$$P_w = \frac{G_a wh}{33000}$$

where w = weight of one gallon of the liquid in pounds and h = total head in ft.

62. The **indicated horsepower** of the steam or liquid end of a pump can be calculated by the well-known formula

$$P_i = \frac{plan}{33,000}$$

where

p = indicated mean effective pressure in pounds per square inch in either steam or liquid cylinder

l = length of stroke in feet

a = area of steam or liquid piston in square inches

n = number of effective strokes per minute

To avoid errors use the following relative values for n: Simple single-acting, n = r.p.m.; simplex double-acting, n = 2 × r.p.m.; Duplex double-acting, n = 4 × r.p.m.

63. This horsepower is calculated from actual or theoretical indicator diagrams.

64. The term **brake horsepower** (P_b) is used only in connection with pumps driven from motors, steam turbines or other prime movers and is the power actually transmitted to the pump by its prime mover. In direct-acting pumps P_b horsepower cannot be measured and therefore has no meaning. In hydraulic practice the term is applied to two classes of pumps centrifugal and power-driven displacement pumps. Brake horsepower is expressed

$$P_b = \frac{P_w}{\mu_m}$$

65. Indicators and Indicator Diagrams.—Indicator cards or diagrams are quite as useful in their applications to steam pumps as to steam engines. Many-power plant and marine engineers who indicate their main engines at regular intervals rarely think of applying the same useful instrument to their steam pumps. An indicator diagram is the result of two movements: a horizontal movement of the paper in exact correspondence with the movement of the piston, and a vertical movement of the pencil in exact ratio to the pressure exerted in the cylinder of the pump. It represents by its length the stroke of the pump, by its height at any point the pressure on the piston at the corresponding point in the stroke, and by its area the work done in the cylinder. A diagram shows the pressure acting on one side of the piston only,

during both the forward and return stroke. To show the corresponding pressures on the other side of the piston another diagram must be taken from the other end of the cylinder. By using a three-way cock the diagram from both ends of the cylinder may be taken on the same paper.

66. An indicator diagram must be carefully analyzed and interpreted or the results obtained from it will be of little or no value. The indicator diagram will show the quantity of power developed in the cylinder and the quantity lost in various ways; by wire drawing through the steam ports and liquid valves, by back pressure, by leakage, etc. Taken in conjunction with measurements of feed water and condensation and measurement of exhaust steam with the amount of fuel used, the indicator diagram furnishes many other facts of importance.

67. The main lever of the steam valve gear of a duplex pump affords a convenient reducing motion to which the cord for operating the paper drum of the indicator may be attached. A point having a travel equal to the length of the indicator diagram should be located on this lever. Where this is not possible a special reducing lever, a Brumbo pulley, a pantograph or other form of reducing motion should be employed. Once this point is located it is convenient to drill and tap a small hole in the lever (about 1/8 in.) and insert a small bolt or screw with a "collar-button" head. A loop or eye in the end of the cord can then be readily slipped on or off for connecting or disconnecting the drum motion. For all practical purposes one indicator can be used for both the steam end and the liquid end. The connections to the cylinders should be made as short and direct as possible, using pipes not less than ½-in. diameter. If long pipes are necessary they should be larger than ½-in. and on the steam end they should be covered with a non-conducting material. For attaching the indicator a three-way cock should be located midway in the line of pipe connecting the holes at either end of the cylinder. By this arrangement diagrams can be taken from either end by simply turning the handle of the three-way cock.

68. To save time both the **head-end** and **cradle-end** diagrams can be taken on the same paper, provided they are not confused with each other afterwards; and to prevent this, they should be care-

fully marked at the time they are taken. In actually taking cards from any pump, however, take them from both ends of all cylinders, as there is no better way of ascertaining whether or not the pump is working correctly and doing equal amounts of work at each end of each stroke.

69. It should be noted that we are using the expression "head" and "cradle" ends for the cylinders to correspond with head end and crank end as generally used on a horizontal steam end. For a vertical pump the expression "top end" and "bottom end" may be used if desired, or "head end" and "cradle end."

70. In **taking an indicator diagram,** always apply pencil to the card for at least one stroke with the cock in neutral position. This will draw a straight line on the card called the "atmosphericline," as under the conditions given there is only the pressure of the atmosphere acting on the indicator piston. After drawing the atmospheric line, throw the cock open and take the diagram proper.

71. The **indicated horsepower** of either the steam end or liquid end of a pump may be computed from the diagram by formula:

$$\frac{plan}{33.000} \text{ (see par. 62).}$$

To find the mean effective pressure p multiply the average height in inches of diagram by the scale of the spring used in taking the diagram and the result will be the mean effective pressure in pounds per square inch. The area of the diagram is measured by a planimeter or by dividing it into equal parts, usually ten, and measuring their average pressure or heights. Of the two methods the planimeter is the quicker and more accurate. The length of the stroke, l, should be taken from actual measurement at the time the diagrams are taken, as a pump rated at 24-in. stroke, for instance, might not be making this exact stroke at the time. It is possible with practically all makes of steam pumps to adjust the steam valves so as to vary the stroke; hence the need for making an accurate measurement of the stroke at the time of indicating the pump. The area a is calculated from the diameter of the piston or plunger. The number of revolutions n must also be taken at the time the diagram is made. The area of the diagram in square inches divided by its length in inches gives the average height in inches.

72. Fig. 5 is a head end steam diagram, spring scale 100 lb. The atmospheric line is 5 lb. below the bottom of the card, indicating that much back pressure on the exhaust and showing that the pump is exhausting either into a heater or through a long and possibly small diameter pipe into the open air.

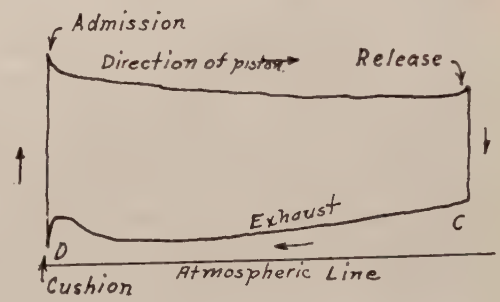

Fig. 5. Head-end, steam-cylinder indicator diagram; spring scale 100 pounds.

73. In Fig. 6 below AB is the atmospheric line of an ordinary card taken from the water end of a pump. The suction stroke starts at M and the line MN is traced while the plunger is "drawing" on the water. Hence this line will lie below the atmospheric line, as the pressure in the pump chamber must be lower than the atmospheric pressure, in order that the latter may force the water in the suction well up into the pump. As water is heavy it has considerable inertia, and in starting to flow into the pump its action is apt to be more or less jerky. This causes the fluctuations in the suction line MN, even though in all properly installed pumps there should be an air chamber on both suction and delivery sides, to smooth out the fluctuations in pressure due to the sudden changes of speed in an incompressible fluid.

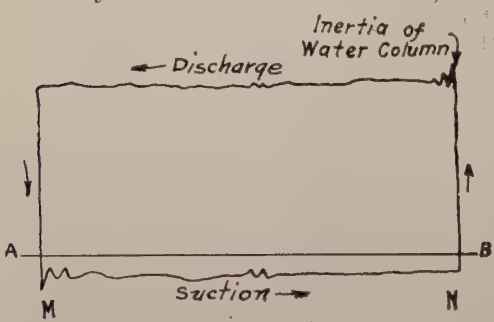

Fig. 6. Indicator card from water end of pump; spring scale 100 pounds.

74. For close **comparison of steam-end and liquid-end diagrams,** they must be reduced to the same scales of stroke and pressure. The vertical ordinates must represent **total** pressures and not pressures in pounds per square inch.

75. Fig. 7 shows the two cards superimposed with the vertical dimensions converted into total pressures on pistons instead of pressures in pounds per square inch. The bottom lines $D\,C$ of the steam card and $M\,N$ of the liquid card are made to coincide (OO_1) as nearly as possible and the atmospheric lines are dispensed with, as it is now effective pressures only that we are interested in and not gage pressures. On the combined diagram the area $A\ B\,H\,G$, which is the excess of the steam-card area over the liquid-card area, represents the "lost work" used up in pump friction.

76. Steam.—In par. 55 the thermal efficiency of a pump is defined as the percentage of all the heat actually used by the machine which appears as useful work. In order to **compute the thermal efficiency** it is necessary to know the volume in cubic feet, the weight and the heat in British thermal units of the steam used. The volume is computed from the size of the steam cylinder. The weight and heat in British thermal units of the steam are taken from the steam tables.

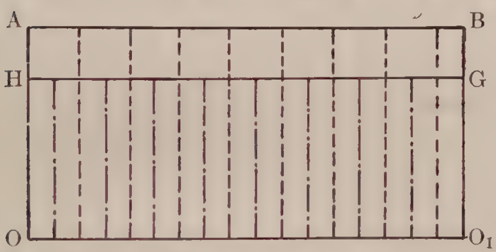

FIG. 7. Indicator cards with vertical dimensions showing total pressure on piston instead of pressures in pounds per square inch.

77. Steam is water vapor. **Saturated steam** is water vapor in the condition in which it is generated from the water with which it is in contact, or it may be described as steam at the maximum pressure and density possible at its temperature. Saturated steam cannot have its temperature lowered without a lowering of its pressure, any loss of heat being made up by the latent heat of such portion as will be condensed. The temperature of saturated steam cannot be increased except when accompanied by a corresponding increase in pressure, any added heat being expended in the evaporation into steam of a portion of the water with which it is in contact.

78. Dry saturated steam contains no water. Saturated steam is

said to be **wet** when it is accompanied by water which is carried along with it, either in the form of a spray or blown along the surface of the steam pipe. If in a mixture of 100 lb. of steam and water there is one pound of water the steam will contain 1 per cent moisture, or is said to be 99 per cent dry saturated steam.

79. Heat may be added to steam not in contact with water, such an addition of heat resulting in an increase of temperature and volume if the pressure remains constant. Steam, the temperature of which thus exceeds that of saturated steam at a corresponding pressure, is said to be **superheated steam.** The properties of superheated steam approximate those of a perfect gas rather than a vapor. Just so long as the temperature of steam is above that of saturated steam at a corresponding pressure it is superheated, and before condensation can take place that superheat must first be lost through radiation or some other means.

80. Table, par. 87, gives the properties of saturated steam and table, par. 88, gives such properties of superheated steam as are necessary in ordinary engineering practice. These tables are based on those computed by Lionel S. Marks and Harvey N. Davis, these being generally accepted as the most correct.

81. Values Used in Steam Tables.—The pressures given in the tables are in pounds per square inch absolute, = gage pressure plus 14.7 lb. per sq. in. (atmosphere).

82. The specific volume of saturated steam at any pressure is the volume in cubic feet of one pound of steam at that pressure.

83. The heat of the liquid is the heat necessary to raise one pound of water from 32 deg. F. to the point of ebullition.

84. The latent heat of evaporation is the sum of the inner latent heat or the heat absorbed during ebullition and the outer latent heat or the heat necessary to overcome the resistance to the increase in volume.

85. The total heat of the steam is the sum of the heat of the liquid and the latent heat of evaporation.

86. The portions of the tables, par. 87 and par. 88, dealing with low-pressure steam, will be of value to engineers in making calculations relating to condensers, heating problems and other low-pressure work.

87. PROPERTIES OF SATURATED STEAM

Reproduced by permission from Marks & Davis. "Steam Tables and Diagrams"

1 Pressure absolute lb. per sq. in.	2 Temperature Deg. F	3 Spec. vol. cu. ft. per lb.	4 Density lb. per Cu. ft.	5 Heat of the liquid B. t. u.	6 Latent heat of evap. B. t. u.	7 Total heat of steam B. t. u.	8 Pressure absolute lb. per sq. in.
p	t	v or s	1/v	h or q	L or r	H	p
1	101.83	333.0	0.00300	69.8	1034.6	1104.4	1
2	126.15	173.5	0.00576	94.0	1021.0	1115.0	2
3	141.52	118.5	0.00845	109.4	1012.3	1121.6	3
4	153.01	90.5	0.01107	120.9	1005.7	1126.5	4
5	162.28	73.33	0.01364	130.1	1000.3	1130.5	5
6	170.06	61.89	0.01616	137.9	995.8	1133.7	6
7	176.85	53.56	0.01867	144.7	991.8	1136.5	7
8	182.86	47.27	0.02115	150.8	988.2	1139.0	8
9	188.27	42.36	0.02361	156.2	985.0	1141.1	9
10	193.22	38.38	0.02606	161.1	982.0	1143.1	10
11	197.75	35.10	0.02849	165.7	979.2	1144.9	11
12	201.96	32.36	0.03090	169.9	976.6	1146.5	12
13	205.87	30.03	0.03330	173.8	974.2	1148.0	13
14	209.55	28.02	0.03569	177.5	971.9	1149.4	14
14.7	212.00	26.79	0.03732	180.0	970.4	1150.4	14.7
15	213.0	26.27	0.03806	181.0	969.7	1150.7	15
16	216.3	24.79	0.04042	184.4	967.6	1152.0	16
17	219.4	23.38	0.04277	187.5	965.6	1153.1	17
18	222.4	22.16	0.04512	190.5	963.7	1154.2	18
19	225.2	21.07	0.04746	193.4	961.8	1155.2	19
20	228.0	20.08	0.04980	196.1	960.0	1156.2	20
21	230.6	19.18	0.05213	198.8	958.3	1157.1	21
22	233.1	18.37	0.05445	201.3	956.7	1158.0	22
23	235.5	17.62	0.05676	203.8	955.1	1158.8	23
24	237.8	16.93	0.05907	206.1	953.5	1159.6	24
25	240.1	16.30	0.0614	208.4	952.0	1160.4	25
26	242.2	15.72	0.0636	210.6	950.6	1161.2	26
27	244.4	15.18	0.0659	212.7	949.2	1161.9	27

87. PROPERTIES OF SATURATED STEAM—Continued

1	2	3	4	5	6	7	8
p	t	v or s	1/v	h or q	L or r	H	p
28	246.4	14.67	0.0682	214.8	947.8	1162.6	28
29	248.4	14.19	0.0705	216.8	946.4	1163.2	29
30	250.3	13.74	0.0728	218.8	945.1	1163.9	30
31	252.2	13.32	0.0751	220.7	943.8	1164.5	31
32	254.1	12.93	0.0773	222.6	942.5	1165.1	32
33	255.8	12.57	0.0795	224.4	941.3	1165.7	33
34	257.6	12.22	0.0818	226.2	940.1	1166.3	34
35	259.3	11.89	0.0841	227.9	938.9	1166.8	35
36	261.0	11.58	0.0863	229.6	937.7	1167.3	36
37	262.6	11.29	0.0886	231.3	936.6	1167.8	37
38	264.2	11.01	0.0908	232.9	935.5	1168.4	38
39	265.8	10.74	0.0931	234.5	934.4	1168.9	39
40	267.3	10.49	0.0953	236.1	933.3	1169.4	40
41	268.7	10.25	0.0976	237.6	932.2	1169.8	41
42	270.2	10.02	0.0998	239.1	931.2	1170.3	42
43	271.7	9.80	0.1020	240.5	930.2	1170.7	43
44	273.1	9.59	0.1043	242.0	929.2	1171.2	44
45	274.5	9.39	0.1065	243.4	928.2	1171.6	45
46	275.8	9.20	0.1087	244.8	927.2	1172.0	46
47	277.2	9.02	0.1109	246.1	926.3	1172.4	47
48	278.5	8.84	0.1131	247.5	925.3	1172.8	48
49	279.8	8.67	0.1153	248.8	924.4	1173.2	49
50	281.0	8.51	0.1175	250.1	923.5	1173.6	50
51	282.3	8.35	0.1197	251.4	922.6	1174.0	51
52	283.5	8.20	0.1219	252.6	921.7	1174.3	52
53	284.7	8.05	0.1241	253.9	920.8	1174.7	53
54	285.9	7.91	0.1263	255.1	919.9	1175.0	54
55	287.1	7.78	0.1285	256.3	919.0	1175.4	55
56	288.2	7.65	0.1307	257.5	918.2	1175.7	56
57	289.4	7.52	0.1329	258.7	917.4	1176.0	57

87. PROPERTIES OF SATURATED STEAM—Continued

1	2	3	4	5	6	7	8
p	t	v or s	1/v	h or q	L or r	H	p
58	290.5	7.40	0.1350	259.8	916.5	1176.4	58
59	291.6	7.28	0.1372	261.0	915.7	1176.7	59
60	292.7	7.17	0.1394	262.1	914.9	1177.0	60
61	293.8	7.06	0.1416	263.2	914.1	1177.3	61
62	294.9	6.95	0.1438	264.3	913.3	1177.6	62
63	295.9	6.85	0.1460	265.4	912.5	1177.9	63
64	297.0	6.75	0.1482	266.4	911.8	1178.2	64
65	298.0	6.65	0.1503	267.5	911.0	1178.5	65
66	299.0	6.56	0.1525	268.5	910.2	1178.8	66
67	300.0	6.47	0.1547	269.6	909.5	1179.0	67
68	301.0	6.38	0.1569	270.6	908.7	1179.3	68
69	302.0	6.29	0.1590	271.6	908.0	1179.6	69
70	302.9	6.20	0.1612	272.6	907.2	1179.8	70
71	303.9	6.12	0.1634	273.6	906.5	1180.1	71
72	304.8	6.04	0.1656	274.5	905.8	1180.4	72
73	305.8	5.96	0.1678	275.5	905.1	1180.6	73
74	306.7	5.89	0.1699	276.5	904.4	1180.9	74
75	307.6	5.81	0.1721	277.4	903.7	1181.1	75
76	308.5	5.74	0.1743	278.3	903.0	1181.4	76
77	309.4	5.67	0.1764	279.3	902.3	1181.6	77
78	310.3	5.60	0.1786	280.2	901.7	1181.8	78
79	311.2	5.54	0.1808	281.1	901.0	1182.1	79
80	312.0	5.47	0.1829	282.0	900.3	1182.3	80
81	312.9	5.41	0.1851	282.9	899.7	1182.5	81
82	313.8	5.34	0.1873	283.8	899.0	1182.8	82
83	314.6	5.28	0.1894	284.6	898.4	1183.0	83
84	315.4	5.22	0.1915	285.5	897.7	1183.2	84
85	316.3	5.16	0.1937	286.3	897.1	1183.4	85
86	317.1	5.10	0.1959	287.2	896.4	1183.6	86
87	317.9	5.05	0.1980	288.0	895.8	1183.8	87

87. PROPERTIES OF SATURATED STEAM—Continued

1	2	3	4	5	6	7	8
p	t	v or s	1/v	h or q	L or r	H	p
88	318.7	5.00	0.2001	288.9	895.2	1184.0	88
89	319.5	4.94	0.2023	289.7	894.6	1184.2	89
90	320.3	4.89	0.2044	290.5	893.9	1184.4	90
91	321.1	4.84	0.2065	291.3	893.3	1184.6	91
92	321.8	4.79	0.2087	292.1	892.7	1184.8	92
93	322.6	4.74	0.2109	292.9	892.1	1185.0	93
94	323.4	4.69	0.2130	293.7	891.5	1185.2	94
95	324.1	4.65	0.2151	294.5	890.9	1185.4	95
96	324.9	4.60	0.2172	295.3	890.3	1185.6	96
97	325.6	4.56	0.2193	296.1	889.7	1185.8	97
98	326.4	4.51	0.2215	296.8	889.2	1186.0	98
99	327.1	4.47	0.2237	297.6	888.6	1186.2	99
100	327.8	4.429	0.2258	298.3	888.0	1186.3	100
101	328.6	4.388	0.2279	299.1	887.4	1186.5	101
102	329.3	4.347	0.2300	299.8	886.9	1186.7	102
103	330.0	4.307	0.2322	300.6	886.3	1186.9	103
104	330.7	4.268	0.2343	301.3	885.8	1187.0	104
105	331.4	4.230	0.2365	302.0	885.2	1187.2	105
106	332.0	4.192	0.2386	302.7	884.7	1187.4	106
107	332.7	4.155	0.2408	303.4	884.1	1187.5	107
108	333.4	4.118	0.2429	304.1	883.6	1187.7	108
109	334.1	4.082	0.2450	304.8	883.0	1187.9	109
110	334.8	4.047	0.2472	305.5	882.5	1188.0	110
111	335.4	4.012	0.2493	306.2	881.9	1188.2	111
112	336.1	3.978	0.2514	306.9	881.4	1188.4	112
113	336.8	3.945	0.2535	307.6	880.9	1188.5	113
114	337.4	3.912	0.2556	308.3	880.4	1188.7	114
115	338.1	3.880	0.2577	309.0	879.8	1188.8	115
116	338.7	3.848	0.2599	309.6	879.3	1189.0	116
117	339.4	3.817	0.2620	310.3	878.8	1189.1	117

87. PROPERTIES OF SATURATED STEAM—Continued

1	2	3	4	5	6	7	8
p	t	v or s	1/v	h or q	L or r	H	p
118	340.0	3.786	0.2641	311.0	878.3	1189.3	118
119	340.6	3.756	0.2662	311.6	877.8	1189.4	119
120	341.3	3.726	0.2683	312.3	877.2	1189.6	120
121	341.9	3.697	0.2705	313.0	876.7	1189.7	121
122	342.5	3.668	0.2726	313.6	876.2	1189.8	122
123	343.2	3.639	0.2748	314.3	875.7	1190.0	123
124	343.8	3.611	0.2769	314.9	875.2	1190.1	124
125	344.4	3.583	0.2791	315.5	874.7	1190.3	125
126	345.0	3.556	0.2812	316.2	874.2	1190.4	126
127	345.6	3.530	0.2833	316.8	873.8	1190.5	127
128	346.2	3.504	0.2854	317.4	873.3	1190.7	128
129	346.8	3.478	0.2875	318.0	872.8	1190.8	129
130	347.4	3.452	0.2897	318.6	872.3	1191.0	130
131	348.0	3.427	0.2918	319.3	871.8	1191.1	131
132	348.5	3.402	0.2939	319.9	871.3	1191.2	132
133	349.1	3.378	0.2960	320.5	870.9	1191.3	133
134	349.7	3.354	0.2981	321.1	870.4	1191.5	134
135	350.3	3.331	0.3002	321.7	869.9	1191.6	135
136	350.8	3.308	0.3023	322.3	869.4	1191.7	136
137	351.4	3.285	0.3044	322.8	869.0	1191.8	137
138	352.0	3.263	0.3065	323.4	868.5	1192.0	138
139	352.5	3.241	0.3086	324.0	868.1	1192.1	139
140	353.1	3.219	0.3107	324.6	867.6	1192.2	140
141	353.6	3.197	0.3129	325.2	867.2	1192.3	141
142	354.2	3.175	0.3150	325.8	866.7	1192.5	142
143	354.7	3.154	0.3171	326.3	866.3	1192.6	143
144	355.3	3.133	0.3192	326.9	865.8	1192.7	144
145	355.8	3.112	0.3213	327.4	865.4	1192.8	145
146	356.3	3.092	0.3234	328.0	864.9	1192.9	146
147	356.9	3.072	0.3255	328.6	864.5	1193.0	147

87. PROPERTIES OF SATURATED STEAM—Continued

1	2	3	4	5	6	7	8
p	t	v or s	1/v	h or q	L or r	H	p
148	357.4	3.052	0.3276	329.1	864.0	1193.2	148
149	357.9	3.033	0.3297	329.7	863.6	1193.3	149
150	358.5	3.012	0.3320	330.2	863.2	1193.4	150
151	359.0	2.993	0.3341	330.8	862.7	1193.5	151
152	359.5	2.974	0.3362	331.4	862.3	1193.6	152
153	360.0	2.956	0.3383	331.9	861.8	1193.7	153
154	360.5	2.938	0.3404	332.4	861.4	1193.8	154
155	361.0	2.920	0.3425	332.9	861.0	1194.0	155
156	361.6	2.902	0.3446	333.5	860.6	1194.1	156
157	362.1	2.885	0.3467	334.0	860.1	1194.2	157
158	362.6	2.868	0.3488	334.6	859.7	1194.3	158
159	363.1	2.851	0.3508	335.1	859.3	1194.4	159
160	363.6	2.834	0.3529	335.6	858.8	1194.5	160
161	364.1	2.818	0.3549	336.2	858.4	1194.6	161
162	364.6	2.801	0.3570	336.7	858.0	1194.7	162
163	365.1	2.785	0.3591	337.2	857.6	1194.8	163
164	365.6	2.769	0.3612	337.7	857.2	1194.9	164
165	366.0	2.753	0.3633	338.2	856.8	1195.0	165
166	366.5	2.737	0.3654	338.7	856.4	1195.1	166
167	367.0	2.721	0.3675	339.2	855.9	1195.2	167
168	367.5	2.706	0.3696	339.7	855.5	1195.3	168
169	368.0	2.690	0.3717	340.2	855.1	1195.4	169
170	368.5	2.675	0.3738	340.7	854.7	1195.4	170
171	368.9	2.660	0.3759	341.2	854.3	1195.5	171
172	369.4	2.645	0.3780	341.7	853.9	1195.6	172
173	369.9	2.631	0.3801	342.2	853.5	1195.7	173
174	370.4	2.616	0.3822	342.7	853.1	1195.8	174
175	370.8	2.602	0.3843	343.2	852.7	1195.9	175
176	371.3	2.588	0.3864	343.7	852.3	1196.0	176
177	371.7	2.574	0.3885	344.2	851.9	1196.1	177

87. PROPERTIES OF SATURATED STEAM—Continued

1	2	3	4	5	6	7	8
p	t	v or s	1/v	h or q	L or r	H	p
178	372.2	2.560	0.3906	344.7	851.5	1196.2	178
179	372.7	2.547	0.3927	345.2	851.2	1196.3	179
180	373.1	2.533	0.3948	345.6	850.8	1196.4	180
181	373.6	2.520	0.3969	346.1	850.4	1196.5	181
182	374.0	2.507	0.3989	346.6	850.0	1196.6	182
183	374.5	2.494	0.4010	347.1	849.6	1196.7	183
184	374.9	2.481	0.4031	347.6	849.2	1196.8	184
185	375.4	2.468	0.4052	348.0	848.8	1196.8	185
186	375.8	2.455	0.4073	348.5	848.4	1196.9	186
187	376.3	2.443	0.4094	349.0	848.0	1197.0	187
188	376.7	2.430	0.4115	349.4	847.7	1197.1	188
189	377.2	2.418	0.4136	349.9	847.3	1197.2	189
190	377.6	2.406	0.4157	350.4	846.9	1197.3	190
191	378.0	2.393	0.4178	350.8	846.5	1197.3	191
192	378.5	2.381	0.4199	351.3	846.1	1197.4	192
193	378.9	2.369	0.4220	351.7	845.8	1197.5	193
194	379.3	2.358	0.4241	352.2	845.4	1197.6	194
195	379.8	2.346	0.4262	352.7	845.0	1197.7	195
196	380.2	2.335	0.4283	353.1	844.7	1197.8	196
197	380.6	2.323	0.4304	353.6	844.3	1197.8	197
198	381.0	2.312	0.4325	354.0	843.9	1197.9	198
199	381.4	2.301	0.4346	354.4	843.6	1198.0	199
200	381.9	2.290	0.437	354.9	843.2	1198.1	200
201	382.3	2.279	0.439	355.3	842.8	1198.2	201
202	382.7	2.269	0.441	355.8	842.4	1198.2	202
203	383.1	2.258	0.443	356.2	842.1	1198.3	203
204	383.5	2.247	0.445	356.7	841.7	1198.4	204
205	384.0	2.237	0.447	357.1	841.4	1198.5	205
206	384.4	2.227	0.449	357.5	841.0	1198.5	206
207	384.8	2.217	0.451	358.0	840.6	1198.6	207

87. PROPERTIES OF SATURATED STEAM—Continued

1	2	3	4	5	6	7	8
p	t	v or s	1/v	h or q	L or r	H	p
208	385.2	2.207	0.453	358.4	840.3	1198.7	208
209	385.6	2.197	0.455	358.8	839.9	1198.8	209
210	386.0	2.187	0.457	359.2	839.6	1198.8	210
211	386.4	2.177	0.459	359.6	839.3	1198.9	211
212	386.8	2.167	0.461	360.1	838.9	1199.0	212
213	387.2	2.158	0.463	360.5	838.6	1199.1	213
214	387.6	2.148	0.466	360.9	838.2	1199.1	214
215	388.0	2.138	0.468	361.4	837.9	1199.2	215
216	388.4	2.128	0.470	361.8	837.5	1199.3	216
217	388.8	2.118	0.472	362.2	837.2	1199.4	217
218	389.1	2.109	0.474	362.6	836.8	1199.4	218
219	389.5	2.100	0.476	363.0	836.5	1199.5	219
220	389.9	2.091	0.478	363.4	836.2	1199.6	220
221	390.3	2.082	0.480	363.8	835.8	1199.6	221
222	390.7	2.073	0.482	364.2	835.5	1199.7	222
223	391.1	2.064	0.485	364.6	835.1	1199.8	223
224	391.5	2.055	0.487	365.0	834.8	1199.8	224
225	391.9	2.046	0.489	365.5	834.4	1199.9	225
226	392.2	2.038	0.491	365.9	834.1	1200.0	226
227	392.6	2.030	0.493	366.3	833.8	1200.0	227
228	393.0	2.021	0.495	366.7	833.4	1200.1	228
229	393.4	2.013	0.497	367.1	833.1	1200.2	229
230	393.8	2.004	0.499	367.5	832.8	1200.2	230
231	394.1	1.996	0.501	367.9	832.4	1200.3	231
232	394.5	1.988	0.503	368.3	832.1	1200.4	232
233	394.9	1.980	0.505	368.7	831.8	1200.4	233
234	395.2	1.972	0.507	369.0	831.4	1200.5	234
235	395.6	1.964	0.509	369.4	831.1	1200.6	235
236	396.0	1.956	0 511	369.8	830.8	1200.6	236
237	396.4	1.948	0.513	370.2	830.4	1200.7	237

87. PROPERTIES OF SATURATED STEAM—Continued

1	2	3	4	5	6	7	8
p	t	v or s	1/v	h or q	L or r	H	p
238	396.7	1.940	0.515	370.6	830.1	1200.7	238
239	397.1	1.932	0.518	371.0	829.8	1200.8	239
240	397.4	1.924	0.520	371.4	829.5	1200.9	240
241	397.8	1.917	0.522	371.8	829.2	1200.9	241
242	398.2	1.909	0.524	372.2	828.8	1201.0	242
243	398.5	1.902	0.526	372.6	828.5	1201.1	243
244	398.9	1.894	0.528	372.9	828.2	1201.1	244
245	399.3	1.887	0.530	373.3	827.9	1201.2	245
246	399.6	1.879	0.532	373.7	827.5	1201.2	246
247	400.0	1.872	0.534	374.1	827.2	1201.3	247
248	400.3	1.864	0.536	374.5	826.9	1201.4	248
249	400.7	1.857	0.538	374.8	826.6	1201.4	249
250	401.1	1.850	0.541	375.2	826.3	1201.5	250
252	401.8	1.836	0.545	376.0	825.6	1201.6	252
254	402.4	1.822	0.549	376.7	825.0	1201.7	254
256	403.1	1.809	0.553	377.5	824.4	1201.8	256
258	403.8	1.795	0.557	378.2	823.7	1201.9	258
260	404.5	1.782	0.561	378.9	823.1	1202.1	260
262	405.2	1.769	0.565	379.6	822.5	1202.2	262
264	405.9	1.756	0.569	380.4	821.9	1202.3	264
266	406.6	1.743	0.574	381.1	821.3	1202.4	266
268	407.2	1.731	0.578	381.8	820.7	1202.5	268
270	407.9	1.718	0.582	382.5	820.1	1202.6	270
272	408.6	1.705	0.587	383.2	819.5	1202.7	272
274	409.2	1.693	0.591	383.9	818.9	1202.8	274
276	409.9	1.681	0.595	384.6	818.3	1202.9	276
278	410.5	1.669	0.599	385.3	817.7	1203.0	278
280	411.2	1.658	0.603	386.0	817.1	1203.1	280
282	411.8	1.646	0.608	386.7	816.5	1203.2	282
284	412.4	1.635	0.612	387.4	815.9	1203.3	284

87. PROPERTIES OF SATURATED STEAM—Concluded

1	2	3	4	5	6	7	8
p	t	v or s	1/v	h or q	L or r	H	p
286	413.1	1.624	0.616	388.1	815.4	1203.4	286
288	413.7	1.613	0.620	388.7	814.8	1203.5	288
290	414.4	1.602	0.624	389.4	814.2	1203.6	290
292	415.0	1.591	0.629	390.1	813.6	1203.7	292
294	415.6	1.581	0.633	390.8	813.0	1203.8	294
296	416.2	1.571	0.637	391.4	812.5	1203.9	296
298	416.8	1.561	0.641	392.1	811.9	1204.0	298
300	417.5	1.551	0.645	392.7	811.3	1204.1	300
310	420.5	1.502	0.666	395.9	808.5	1204.5	310
320	423.4	1.456	0.687	399.1	805.8	1204.9	320
330	426.3	1.413	0.708	402.2	803.1	1205.3	330
340	429.1	1.372	0.729	405.3	800.4	1205.7	340
350	431.9	1.334	0.750	408.2	797.8	1206.1	350
360	434.6	1.298	0.770	411.2	795.3	1206.4	360
370	437.2	1.264	0.791	414.0	792.8	1206.8	370
380	439.8	1.231	0.812	416.8	790.3	1207.1	380
390	442.3	1.200	0.833	419.5	787.9	1207.4	390
400	444.7	1.17	0.86	422.	786.	1208.	400
410	447.2	1.14	0.88	425.	783.	1208.	410
420	449.6	1.11	0.90	427.	780.	1208.	420
430	451.9	1.09	0.92	430.	778.	1208.	430
440	454.2	1.06	0.94	433.	776.	1208.	440
450	456.5	1.04	0.96	435.	774.	1209.	450
460	458.7	1.01	0.99	438.	771.	1209.	460
470	460.9	0.99	1.01	440.	769.	1209.	470
480	463.0	0.97	1.03	443.	767.	1209.	480
490	465.1	0.95	1.05	445.	764.	1210.	490
500	467.2	0.93	1.08	448.	762.	1210.	500
525	472.3	0.89	1.12	453.	757.	1210.	525
550	477.2	0.85	1.18	458.	752.	1210.	550
575	481.9	0.81	1.24	464.	747.	1211.	575
600	486.4	0.781	1.28	469.	742.	1211.	600

88. PROPERTIES OF SUPERHEATED STEAM

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Pressure Pounds Absolute		Saturated Steam	Degrees of Superheat					
			50	100	150	200	250	300
5	t	162.3	212.3	262.3	312.3	362.3	412.3	462.3
	v	73.3	79.7	85.7	91.8	97.8	103.8	109.8
	h	1130.5	1153.5	1176.4	1199.5	1222.5	1245.6	1268.7
10	t	193.2	243.2	293.2	343.2	393.2	443.2	493.2
	v	38.4	41.5	44.6	47.7	50.7	53.7	56.7
	h	1143.1	1166.3	1189.5	1212.7	1236.0	1259.3	1282.5
15	t	213.0	263.0	313.0	363.0	413.0	463.0	513.0
	v	26.27	28.40	30.46	32.50	34.53	36.56	38.58
	h	1150.7	1174.2	1197.6	1221.0	1244.4	1267.7	1291.1
20	t	228.0	278.0	328.0	378.0	428.0	478.0	528.0
	v	20.08	21.69	23.25	24.80	26.33	27.85	29.37
	h	1156.2	1179.9	1203.5	1227.1	1250.6	1274.1	1297.6
25	t	240.1	290.1	340.1	390.1	440.1	490.1	540.1
	v	16.30	17.60	18.86	20.10	21.32	22.55	23.77
	h	1160.4	1184.4	1208.2	1231.9	1255.6	1279.2	1302.8
30	t	250.4	300.4	350.4	400.4	450.4	500.4	550.4
	v	13.74	14.83	15.89	16.93	17.97	18.99	20.00
	h	1163.9	1188.1	1212.1	1236.0	1259.7	1283.4	1307.1
35	t	259.3	309.3	359.3	409.3	459.3	509.3	559.3
	v	11.89	12.85	13.75	14.65	15.54	16.42	17.30
	h	1166.8	1191.3	1215.4	1239.4	1263.3	1287.1	1310.8
40	t	267.3	317.3	367.3	417.3	467.3	517.3	567.3
	v	10.49	11.33	12.13	12.93	13.70	14.48	15.25
	h	1169.4	1194.0	1218.4	1242.4	1266.4	1290.3	1314.1
45	t	274.5	324.5	374.5	424.5	474.5	524.5	574.5
	v	9.39	10.14	10.86	11.57	12.27	12.96	13.65
	h	1171.6	1196.6	1221.0	1245.2	1269.3	1293.2	1317.0

t = Temperature, degrees Fahrenheit.
v = Specific volume, in cubic feet per pound.
h = Total heat from water at 32 degrees, B.t.u.

88. PROPERTIES OF SUPERHEATED STEAM—Continued

Pressure Pounds Absolute		Saturated Steam	Degrees of Superheat 50	100	150	200	250	300
	t	281.0	331.0	381.0	431.0	481.0	531.0	581.0
50	v	8.51	9.19	9.84	10.48	11.11	11.74	12.36
	h	1173.6	1198.8	1223.4	1247.7	1271.8	1295.8	1319.7
	t	287.1	337.1	387.1	437.1	487.1	537.1	587.1
55	v	7.78	8.40	9.00	9.59	10.16	10.73	11.30
	h	1175.4	1200.8	1225.6	1250.0	1274.2	1298.1	1322.0
	t	292.7	342.7	392.7	442.7	492.7	542.7	592.7
60	v	7.17	7.75	8.30	8.84	9.36	9.89	10.41
	h	1177.0	1202.6	1227.6	1252.1	1276.4	1300.4	1324.3
	t	298.0	348.0	398.0	448.0	498.0	548.0	598.0
65	v	6.65	7.20	7.70	8.20	8.69	9.17	9.65
	h	1178.5	1204.4	1229.5	1254.0	1278.4	1302.4	1326.4
	t	302.9	352.9	402.9	452.9	502.9	552.9	602.9
70	v	6.20	6.71	7.18	7.65	8.11	8.56	9.01
	h	1179.8	1205.9	1231.2	1255.8	1280.2	1304.3	1328.3
	t	307.6	357.6	407.6	457.6	507.6	557.6	607.6
75	v	5.81	6.28	6.73	7.17	7.60	8.02	8.44
	h	1181.1	1207.5	1232.8	1257.5	1282.0	1306.1	1330.1
	t	312.0	362.0	412.0	462.0	512.0	562.0	612.0
80	v	5.47	5.92	6.34	6.75	7.17	7.56	7.95
	h	1182.3	1208.8	1234.3	1259.0	1283.6	1307.8	1331.9
	t	316.3	366.3	416.3	466.3	516.3	566.3	616.3
85	v	5.16	5.59	5.99	6.38	6.76	7.14	7.51
	h	1183.4	1210.2	1235.8	1260.6	1285.2	1309.4	1333.5
	t	320.3	370.3	420.3	470.3	520.3	570.3	620.3
90	v	4.89	5.29	5.67	6.04	6.40	6.76	7.11
	h	1184.4	1211.4	1237.2	1262.0	1286.6	1310.8	1334.9
	t	324.1	374.1	424.1	474.1	524.1	574.1	624.1
95	v	4.65	5.03	5.39	5.74	6.09	6.43	6.76
	h	1185.4	1212.6	1238.4	1263.4	1288.1	1312.3	1336.4

t = Temperature, degrees Fahrenheit.
v = Specific volume, in cubic feet per pound.
h = Total heat from water at 32 degrees, B.t.u.

88. PROPERTIES OF SUPERHEATED STEAM—Continued

Pressure Pounds Absolute		Saturated Steam	Degrees of Superheat 50	100	150	200	250	300
100	t	327.8	377.8	427.8	477.8	527.8	577.8	627.8
	v	4.43	4.79	5.14	5.47	5.80	6.12	6.44
	h	1186.3	1213.8	1239.7	1264.7	1289.4	1313.6	1337.8
105	t	331.4	381.4	431.4	481.4	531.4	581.4	631.4
	v	4.23	4.58	4.91	5.23	5.54	5.85	6.15
	h	1187.2	1214.9	1240.8	1265.9	1290.6	1314.9	1339.1
110	t	334.8	384.8	434.8	484.8	534.8	584.8	634.8
	v	4.05	4.38	4.70	5.01	5.31	5.61	5.90
	h	1188.0	1215.9	1242.0	1267.1	1291.9	1316.2	1340.4
115	t	338.1	388.1	438.1	488.1	538.1	588.1	638.1
	v	3.88	4.20	4.51	4.81	5.09	5.38	5.66
	h	1188.8	1216.9	1243.1	1268.2	1293.0	1317.3	1341.5
120	t	341.3	391.3	441.3	491.3	541.3	591.3	641.3
	v	3.73	4.04	4.33	4.62	4.89	5.17	5.44
	h	1189.6	1217.9	1244.1	1269.3	1294.1	1318.4	1342.7
125	t	344.4	394.4	444.4	494.4	544.4	594.4	644.4
	v	3.58	3.88	4.17	4.45	4.71	4.97	5.23
	h	1190.3	1218.8	1245.1	1270.4	1295.2	1319.5	1343.8
130	t	347.4	397.4	447.4	497.4	547.4	597.4	647.4
	v	3.45	3.74	4.02	4.28	4.54	4.80	5.05
	h	1191.0	1219.7	1246.1	1271.4	1296.2	1320.6	1344.9
135	t	350.3	400.3	450.3	500.3	550.3	600.3	650.3
	v	3.33	3.61	3.88	4.14	4.38	4.63	4.87
	h	1191.6	1220.6	1247.0	1272.3	1297.2	1321.6	1345.9
140	t	353.1	403.1	453.1	503.1	553.1	603.1	653.1
	v	3.22	3.49	3.75	4.00	4.24	4.48	4.71
	h	1192.2	1221.4	1248.0	1273.3	1298.2	1322.6	1346.9
145	t	355.8	405.8	455.8	505.8	555.8	605.8	655.8
	v	3.12	3.38	3.63	3.87	4.10	4.33	4.56
	h	1192.8	1222.2	1248.8	1274.2	1299.1	1323.6	1347.9

t = Temperature, degrees Fahrenheit.
v = Specific volume, in cubic feet per pound.
h = Total heat from water at 32 degrees, B.t.u.

88. PROPERTIES OF SUPERHEATED STEAM—Continued

Pressure Pounds Absolute		Saturated Steam	Degrees of Superheat 50	100	150	200	250	300
	t	358.5	408.5	458.5	508.5	558.5	608.5	658.5
150	v	3.01	3.27	3.51	3.75	3.97	4.19	4.41
	h	1193.4	1223.0	1249.6	1275.1	1300.0	1324.5	1348.8
	t	361.0	411.0	461.0	511.0	561.0	611.0	661.0
155	v	2.92	3.17	3.41	3.63	3.85	4.06	4.28
	h	1194.0	1223.6	1250.5	1276.0	1300.8	1325.3	1349.7
	t	363.6	413.6	463.6	513.6	563.6	613.6	663.6
160	v	2.83	3.07	3.30	3.53	3.74	3.95	4.15
	h	1194.5	1224.5	1251.3	1276.8	1301.7	1326.2	1350.6
	t	366.0	416.0	466.0	516.0	566.0	616.0	666.0
165	v	2.75	2.99	3.21	3.43	3.64	3.84	4.04
	h	1195.0	1225.2	1252.0	1277.6	1302.5	1327.1	1351.5
	t	368.5	418.5	468.5	518.5	568.5	618.5	668.5
170	v	2.68	2.91	3.12	3.34	3.54	3.73	3.92
	h	1195.4	1225.9	1252.8	1278.4	1303.3	1327.9	1352.3
	t	370.8	420.8	470.8	520.8	570.8	620.8	670.8
175	v	2.60	2.83	3.04	3.24	3.44	3.63	3.82
	h	1195.9	1226.6	1253.6	1279.1	1304.1	1328.7	1353.2
	t	373.1	423.1	473.1	523.1	573.1	623.1	673.1
180	v	2.53	2.75	2.96	3.16	3.35	3.54	3.72
	h	1196.4	1227.2	1254.3	1279.9	1304.8	1329.5	1353.9
	t	375.4	425.4	475.4	525.4	575.4	625.4	675.4
185	v	2.47	2.68	2.89	3.08	3.27	3.45	3.63
	h	1196.8	1227.9	1255.0	1280.6	1305.6	1330.2	1354.7
	t	377.6	427.6	477.6	527.6	577.6	627.6	677.6
190	v	2.41	2.62	2.81	3.00	3.19	3.37	3.55
	h	1197.3	1228.6	1255.7	1281.3	1306.3	1330.9	1355.5
	t	379.8	429.8	479.8	529.8	579.8	629.8	679.8
195	v	2.35	2.55	2.75	2.93	3.11	3.29	3.46
	h	1197.7	1229.2	1256.4	1282.0	1307.0	1331.6	1356.2

t = Temperature, degrees Fahrenheit.
v = Specific volume, in cubic feet per pound.
h = Total heat from water at 32 degrees, B.t.u.

88. PROPERTIES OF SUPERHEATED STEAM—Continued

Pressure Pounds Absolute		Saturated Steam	Degrees of Superheat						
			50	100	150	200	250	300
200	t	381.9	431.9	481.9	531.9	581.9	631.9	681.9
	v	2.29	2.49	2.68	2.86	3.04	3.21	3.38
	h	1198.1	1229.8	1257.1	1282.6	1307.7	1332.4	1357.0
205	t	384.0	434.0	484.0	534.0	584.0	634.0	684.0
	v	2.24	2.44	2.62	2.80	2.97	3.14	3.30
	h	1198.5	1230.4	1257.7	1283.3	1308.3	1333.0	1357.7
210	t	386.0	436.0	486.0	536.0	586.0	636.0	686.0
	v	2.19	2.38	2.56	2.74	2.91	3.07	3.23
	h	1198.8	1231.0	1258.4	1284.0	1309.0	1333.7	1358.4
215	t	388.0	438.0	488.0	538.0	588.0	638.0	688.0
	v	2.14	2.33	2.51	2.68	2.84	3.00	3.16
	h	1199.2	1231.6	1259.0	1284.6	1309.7	1334.4	1359.1
220	t	389.9	439.9	489.9	539.9	589.9	639.9	689.9
	v	2.09	2.28	2.45	2.62	2.78	2.94	3.10
	h	1199.6	1232.2	1259.6	1285.2	1310.3	1335.1	1359.8
225	t	391.9	441.9	491.9	541.9	591.9	641.9	691.9
	v	2.05	2.23	2.40	2.57	2.72	2.88	3.03
	h	1199.9	1232.7	1260.2	1285.9	1310.9	1335.7	1360.3
230	t	393.8	443.8	493.8	543.8	593.8	643.8	693.8
	v	2.00	2.18	2.35	2.51	2.67	2.82	2.97
	h	1200.2	1233.2	1260.7	1286.5	1311.6	1336.3	1361.0
235	t	395.6	445.6	495.6	545.6	595.6	645.6	695.6
	v	1.96	2.14	2.30	2.46	2.62	2.77	2.91
	h	1200.6	1233.8	1261.4	1287.1	1312.2	1337.0	1361.7
240	t	397.4	447.4	497.4	547.4	597.4	647.4	697.4
	v	1.92	2.09	2.26	2.42	2.57	2.71	2.85
	h	1200.9	1234.3	1261.9	1287.6	1312.8	1337.6	1362.3
245	t	399.3	449.3	499.3	549.3	599.3	649.3	699.3
	v	1.89	2.05	2.22	2.37	2.52	2.66	2.80
	h	1201.2	1234.8	1262.5	1288.2	1313.3	1338.2	1362.9

t = Temperature, degrees Fahrenheit.
v = Specific volume, in cubic feet per pound.
h = Total heat from water at 32 degrees, B.t.u.

88. PROPERTIES OF SUPERHEATED STEAM—Concluded

Pressure Pounds Absolute		Saturated Steam	Degrees of Superheat					
			50	100	150	200	250	300
	t	401.0	451.0	501.0	551.0	601.0	651.0	701.0
250	v	1.85	2.02	2.17	2.33	2.47	2.61	2.75
	h	1201.5	1235.4	1263.0	1288.8	1313.9	1338.8	1363.5
	t	402.8	452.8	502.8	552.8	602.8	652.8	702.8
255	v	1.81	1.98	2.14	2.28	2.43	2.56	2.70
	h	1201.8	1235.9	1263.6	1289.3	1314.5	1339.3	1364.1

t = Temperature, degrees Fahrenheit.
v = Specific volume, in cubic feet per pound.
h = Total heat from water at 32 degrees, B.t.u.

100. Measurement of Liquids.—The most accurate methods of measuring the quantity of liquid delivered by a pump in a given time interval are to measure the volume or to weigh the liquid delivered. These methods are commercially possible only where small quantities are to be measured. When large quantities of liquids are to be measured, as in the testing laboratories of manufacturers of large pumping machinery, other methods must be used, such as the weir, the Pitot tube or the Venturi meter. While it is true that these methods do not give absolute accuracy, the percentage of error is negligible for commercial purposes.

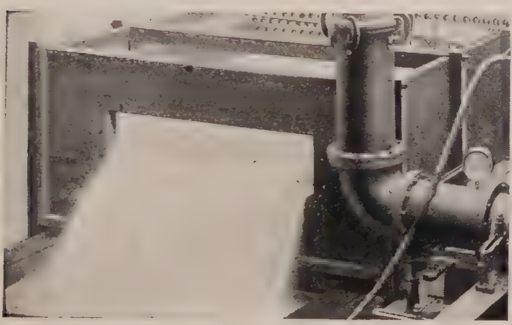


FIG. 8. Weir used for water-flow measurement.

101. A **weir** is a notch in the vertical side of a tank or reservoir through which the water may flow to be measured. The notch is usually rectangular and with a width less than that of the tank, as shown by Fig. 8. To measure the discharge as accurately as possible, the inner edges of the notch shall make a definite angular corner, so that the liquid in flowing out may touch the crest only in a line, thus insuring complete contraction. The crest is the lower edge over which the water passes. For accurate work a thin metal plate is used for a crest. The outer edge of the weir is beveled to an angle of from 30 to 60 degrees. The back side of the weir should be smooth and vertical for a considerable distance downward from the crest.

102. Suitable **baffles** must be built into the tank to break up currents or eddies set up by the discharge of the pump. The liquid should not approach the weir with any noticeable velocity, otherwise a greater quantity would be discharged than indicated by the depth over the crest.

103. The depth of the liquid flowing over the weir is measured

by a **hook gage** (Fig. 9). The hook gage must be located some distance back from the weir in order to avoid the effect of the curvature of the surface of the liquid as it approaches the weir. The hook gage must first be set at zero, at which reading the point of the hook will be level with the crest of the weir. This can be done by a good spirit level or it can be done with the liquid level just at the crest of the weir.

104. A weir with the vertical edges of the notch distant from the sides of the tank or reservoir, so that the sides of the stream may be fully contracted, is known as a weir with **end contractions.**

105. A weir with the vertical edges of the notch coincident with the sides of the tank or reservoir so that the filaments of liquid along the sides pass over the crest without being deflected from the vertical planes in which they move is known as a weir without end contractions.

106. All **weir formulas** do not give the same results. The most widely used is that derived by Mr. James B. Francis from an extensive series of experiments on weirs 10 ft. long. The Francis formulæ for weirs are

$$Q = 3.33\ (L - 0.1\,H)\ H^{3/2} \text{ for one end contracted}$$

$$Q = 3.33\ (L - 0.2\,H)\ H^{3/2} \text{ for both ends contracted}$$

$$Q = 3.33\ L\ H^{3/2} \text{ without end contractions}$$

where Q = cubic feet per second; L = length of weir in feet, and H = head in feet over crest.

107. The **triangular** or **V-notch weir** is frequently used to measure small flows. The 90-deg. triangular notch is the most used. The discharge of a triangular weir may be calculated by the formula

$$Q = 2.544\,H^{5/2}$$

where Q = cubic feet per second, and H = head in feet above apex of triangle.

108. Fig. 10 shows an arrangement for measuring the discharge of a pump by means of a **Pitot tube.** To obtain accurate results by this method it is necessary to use the utmost care in the calibration of the discharge nozzle and the determination of the coefficient C. The Pitot tube measures the velocity head at the

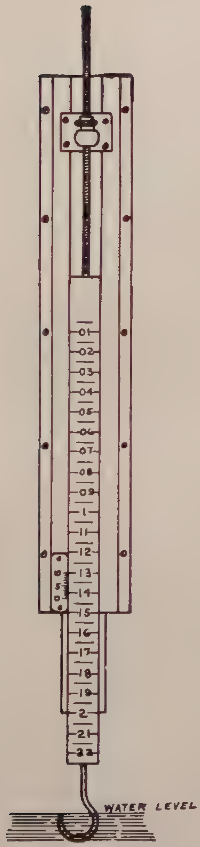


Fig. 9. Hook gage for measuring depth of liquid over weir.

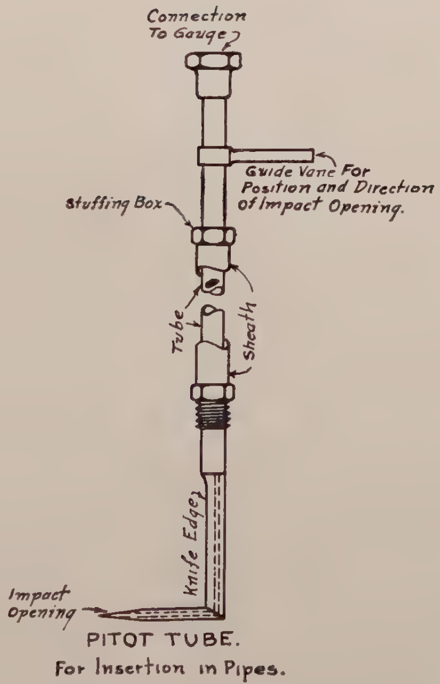


Fig. 10. Pitot tube used for measuring flow of water in pipes.

orifice of the discharge nozzle. The quantity of discharge can be computed from the formula

$$Q = C\ A \sqrt{2\,g\,H}$$

where Q = cubic feet per second; C = coefficient of discharge for the nozzle (0.95 to 0.98); A = area of the nozzle in square feet, and H = the velocity head in feet.

109. Selection of Proper Pump.—In selecting the size and type of pump for a given purpose, a careful analysis should be made of all the factors relating to the installation. Frequently a pump is installed which will do the work but which is nevertheless not the proper type of pump for that particular job. The prospective purchaser of pumping equipment should ask and answer the following questions:

1. What is the maximum amount of liquid the pump is ever apt to be called upon to deliver?
2. What is the nature of the liquid to be pumped? Is it
   - (a) Fresh or salt water, acid, oil, etc.?
   - (b) Thick or thin?
   - (c) Viscous or non-viscous?
   - (d) Cold or hot and the temperature?
   - (e) Clear and free from suspended matter?
   - (f) Dirty or gritty?
   - (g) Does it carry foreign matter in suspension?
   - (h) If so, what is the nature of such foreign matter; is it abrasive or not?
3. What is the maximum discharge pressure the pump must work against?
4. Is the suction (a) lift? or (b) head? and what is the maximum?
5. What is the length of the delivery pipe, and its size, if that has been fixed? 100 ft ~~6~~ Steel
6. Length and diameter of suction pipe?
7. Is the service steady or intermittent?
8. What type of power, if any, is available to drive the pump?
9. What space is available for installing the pump?
10. Will the pump be readily accessible for installation?
11. Is the weight of the pump a matter of serious importance?
12. What are the facilities for transporting the pump from the factory to its final destination?

110. Taking up these questions in order:

111. (1) Maximum amount of liquid the pump is called upon to deliver. This is more important in some cases than in others. Any good pump can be run occasionally for short periods at a capacity considerably in excess of its "rated" capacity, but the purchaser should understand that this emergency **overloading** should be done *only* occasionally and for short periods of time. A reciprocating pump may be operated for short periods at 30 per cent above its rated capacity without causing any damage. Above this point the wear on the valves and other moving parts is excessive and there is always the possibility of damage to the castings from water hammer. Centrifugal pumps that are driven by electric motors are sometimes designed with impellers that cannot be overloaded, so as to prevent possible damage to the motor. With a steam turbine drive it is possible to operate the pump for short periods at overload as high as 25 per cent, without harm to either the pump or the turbine. If any pump is to run even a small part of the time at a load considerably in excess of its mean or average load, then the pump should be specified as of a normal capacity equal to the maximum load.

112. (2) The nature of the liquid has a direct bearing on the type of liquid end and the materials used in its construction.

113. (2a) It is of the utmost importance that the liquid end of a pump be constructed of material best suited for the particular liquid to be handled. Such a list of materials will be found in table, par. 140. For many chemicals, materials can be used, for all parts that come in contact with the liquid, of such a nature that they are affected very little, if at all, by chemical reaction. For other cases, where there is no known material that will long resist the action of the chemicals, the pump can be so constructed that the corroded parts can be easily replaced. There are certain mine waters, for instance, that eat out liquid ends, no matter of what commercially suitable materials they are constructed; for such service, the entire liquid ends are cheaply but substantially constructed, and when eaten out can be replaced at relatively moderate expense.

114. In the case of mine water, it is of no consequence as to the **effect of the pump upon the water,** as it is a waste product any-

how. In chemical plants, fruit and vegetable canneries, pickle factories, etc., where foods are prepared, it is of the greatest importance that the liquids pumped are not contaminated by the pump. It is sometimes necessary to construct a pump with a liquid end of some very expensive alloy or to line or cover all parts coming in contact with the liquid with such an alloy, not so much to protect the pump from the liquid as to protect the liquid from the pump.

115. (2b-c) The type of **valve service** is often determined by the viscosity of the liquid. For thin liquids the standard disk valve is used. For thick liquids and for viscous liquids ball valves, block valves or large diameter double-seat valves are used.

116. (2d) **Temperature.**—In the construction of pumps for handling cold liquids, certain materials are frequently used in rod and piston or plunger packings, and in the valves, which rapidly deteriorate or break down when hot liquids are handled. If a pump is needed for handling hot water or any other hot liquid, the case should always be taken up with the manufacturer. For the effects of temperature on pump suction see par. 119.

117. (2e-f-g-h)—The **condition of the liquid,** i.e., whether it is clear, dirty or contains matter in suspension, also has a bearing on the type of liquid end, and will determine whether a positive-displacement or a centrifugal pump should be used, provided this choice has not already been made on other grounds. Either the piston or plunger type of positive-displacement pumps and the closed-impeller type of centrifugal pump are best suited for clear liquids. The outside-packed plunger pumps and the open-impeller centrifugal pumps are suitable for dirty, gritty liquids. For liquids that carry quantities of foreign matter in suspension always consult the manufacturers, giving full details, and if possible send a sample of the liquid for their inspection.

118. (3) Maximum **discharge pressure** to work against. This is a most important consideration, as it determines the strength of all the liquid-end parts, and taken in conjunction with the steam pressure, if a steam pump is being considered, determines the ratio of size of steam cylinder to size of water cylinder. It is important in considering this question to be sure the maximum

pressure considered is really the maximum, and not merely the static pressure. The total pressure will be the static head, plus the friction head, plus the velocity head. See par. 45.

119. (4) Maximum **suction lift** or maximum **suction head.** If the pump could establish a perfect vacuum, the maximum suction lift for cold water would be 34 ft.; but for all types of pumps it is not good practice to have the suction lift over 20 ft., and not even that much unless the suction pipe is short, free from sharp turns and elbows, and of such diameter that the velocity of the water in it is comparatively low. A mean velocity of 10 ft. per sec. causes a velocity-head of 1.55 ft., from which it can be easily seen that, irrespective of friction, the diameter of a suction pipe should be as large as convenient, and the length as short and direct as possible. The temperature of the liquid also affects the suction and is one of the factors that determines whether the pump can operate with a suction lift or must operate with a positive suction head. For a complete discussion see Sec. VI.

120. (5) Length and size of **delivery pipe.** Here again we desire to keep the velocity as low as convenient, in order to reduce both velocity and friction heads; and for the same reason to use as few sharp turns, fittings, valves, etc., as can be managed. However, this is not of so much importance in the case of the delivery line as in the suction, as a pump can be designed to overcome almost any pressure on the discharge side, while it is strictly limited on the suction end.

121. (6) Length and diameter of suction pipe. This was sufficiently discussed in par. 119.

122. (7) **Steady or intermittent service.** This question was partly answered in par. 111. In general, for intermittent service, a smaller pump may be used than for continuous service, and therefore a cheaper one. It is always the best policy in the long run to install a pump of such size that it will never be called upon to deliver more than 30 per cent in excess of its rated capacity. It is always good practice to install pumps for **intermittent service** of a rather more expensive type than for steady service. A plain iron water cylinder, wet, but not in active use, will rust during periods of idleness, and if not carefully and

regularly inspected, the pump may stick and refuse to work at all when suddenly called upon. Condensed water from the steam line will leak into the steam end of a pump that is standing idle, causing cylinders, pistons and valves to rust and parts which should move freely will stick fast. If a pump is installed for intermittent or emergency service, specially for fire service only, it should be seen to that it is under the care of a competent and reliable man, and that it is inspected and run under full load once every week.

123. (8) **Type of power** available to drive pump. Where steam is available, and especially near or around boilers, the direct-acting pump is cheap, handy, reliable and practically fool-proof. Either simplex or duplex can be used; the duplex being perhaps preferable where the simplest possible apparatus is desired. If the service is such that the centrifugal or turbine pump is preferable, the complete, unit mounted on a common bedplate with steam turbine direct-connected to the pump, is compact, reliable and efficient.

124. Pumps are frequently needed where steam is not available. This is the case in many mines, deep-well and irrigating-pump stations, oil or gasoline-powered ships and boats, hydroelectric power plants and other places too numerous to mention. In such locations a pump can be driven by an electric motor direct-connected or by a belt from a line shaft or other prime mover. The ordinary power pump previously mentioned, with single or double-acting water end, and with one, two or three water cylinders, and pistons or plungers driven from a crankshaft by connecting rods, is preferred by many users. The centrifugal pump can also be driven by any of the means mentioned for power pumps. In mines, the standard steam pump driven by compressed air instead of steam is sometimes used. This is an expensive process, but is justified under certain circumstances.

125. There is an increasing demand for power pumping as a means of water supply for farms, country clubs, and private estates remote from city water supplies. This varies from small automatic or semi-automatic systems for single residences to quite elaborate installations on large estates, dairy farms, and farms

or ranches needing water for irrigation as well as for drinking, sanitary and laundry purposes. The water may be from a creek, a spring or a well. If such a place is handy to an electric power line, the electric drive is undoubtedly the best source of power. If not, the oil or gasoline engine can be used; and occasionally an hydraulic turbine or other water motor. For these private water-supply systems, any type of equipment can be furnished, depending on the circumstances in each individual case; and the Worthington Pump and Machinery Corporation will be pleased to recommend the proper installation upon receipt of inquiry, stating the power available or desired, and the amount of water needed or obtainable. It is always well in such systems to allow for ample storage capacity in a reservoir or overhead tank, located high enough so that the water can flow by gravity to the various taps and outlets desired. This allows a reserve for fire protection, and to carry over a failure in supply during dry weather; and enables repairs or replacement to be made to the machinery without cutting off the water. A float in the tank can be utilized to show the level at any time, and if desired, can be connected up to a low-water alarm, or to an automatic starting and stopping device to the pump motor, if electric drive is used.

126. (9) **Space available for installing pump.** It is sometimes necessary to install a pump in a very limited space, and if a steam pump is wanted, the vertical type is usually preferable. This type is used almost entirely on shipboard, and even when there is plenty of room. Many engineers prefer it. It takes up only a small fraction of the floor space required by a horizontal pump of equal capacity, but should have plenty of space *over* it, where possible. If this space also is limited, the pump can be installed readily enough, but it may be necessary to disconnect it and move it bodily in order to draw pistons, rods, etc. If it is ever necessary to set up a pump in a cramped or limited space, it is well to take up the matter with the Worthington Pump and Machinery Corporation, which will promptly advise whether or not the project is feasible, and what type of pump, if any, can be used.

127. (10) **Accessibility after installation.** Par. 126 bears partly on this topic, and its importance cannot be too strongly ac-

cented. In any new layout, allow for proper pump locations before it is too late. A pump located in an out-of-the-way, or cramped, or dirty and wet, or poorly lighted or dangerous position will surely be neglected by the attendants, will not give satisfactory service, and will be difficult to dismantle and repair. A good steam pump will probably stand more abuse than any other power-actuated machine in use today; but it is entitled to the little attention and consideration that it needs.

128. (11) **Weight** of the pump. In many cases, the weight of the pump is a matter of no special importance, but aboard ships, especially small, light and speedy ones, weight assumes serious importance. It is well to know, then, that it is quite possible to obtain standard pumps of extremely light weight, comparatively speaking, though they are naturally more costly than the heavier ones, heavy castings being replaced by steel forgings, and all metal cut away that can possibly be dispensed with. Such pumps are not better in performance than the heavier ones that are generally used, but their light weight as compared with their capacity makes them readily available in locations where the heavier pumps cannot be used at all.

129. (12) Facilities for **transportation.** Sometimes it is a long journey from the factory where a pump is built to the place where it is needed, and it may be that over a part of this journey the transportation facilities are very poor. Cases sometimes arise where horses or mules must be depended upon, and all freight must be carried on pack saddles. For transportation over such routes, pumps can be supplied that are (a) light in total weight, and (b) that are of sectional construction, so that no single part or section is too heavy for horse or mule back. Such pumps are considerably more expensive than the regular or standard models, but they are well worth the extra cost when the difficulties of transportation are considered.

130. Having gone over in a general and greatly abbreviated way the question of the selection of the right pump for a given job, the prospective buyer, if not already experienced in the selection of pumps, should have a general idea of the problem. But it is strongly urged that if contemplating the purchase of one or more pumps for any purpose whatsoever, the buyer will take

up the question with the engineers of the Worthington Pump and Machinery Corporation, which will be pleased to give correct and up-to-the-minute advice. With many years of experience and research, they will recommend what, in their opinion and judgment, will be the best and most efficient type of installation, and its cost.

FITTINGS FOR PUMPS

Positive-Displacement Pumps.

133. Plain Fitted. Abbreviation P.F. Bronze-lined liquid cylinders, iron water pistons, steel rods, bronze or rubber valves, bronze valve seats, guards and springs.

134. Bronze Fitted. Abbreviation B.F. Bronze-lined liquid cylinders, bronze piston rods, iron water pistons, bronze or rubber valves, bronze valve seats, guards and springs.

135. Full Bronze Fitted. Abbreviation F.B.F. Bronze-lined liquid cylinders, bronze piston rods, bronze water pistons, bronze or rubber valves, bronze valve seats, guards and springs.

Centrifugal Pumps.

136. Plain Fitted. Abbreviation P.F. Steel shaft, iron impellers, bronze impeller bushing rings.

137. Bronze Fitted. Abbreviation B.F. Bronze or bronze-covered steel shaft, bronze impellers, bronze impeller bushing rings.

Special Fittings.

138. All Iron. Abbreviation A.I. All parts of the pump coming in contact with the liquid to be of either cast or malleable iron. Piston rods or shafts of steel.

139. All Bronze. Abbreviation A.B. All parts of the pump coming in contact with the liquid to be of bronze. Piston rods of bronze; shafts to be of bronze or of steel, bronze covered.

139a. Cast Iron Special lined. Abbreviation C.I. followed by name of lining used. Liquid cylinder of cast iron lined with material mentioned.

140. PROPERTIES OF LIQUIDS AND FITTINGS FOR PUMPS

Materials and fittings used in the construction of pumps for handling various liquids. The values in columns 4, 5, 6 and 7 are calculated from data furnished by the Department of Commerce, Bureau of Standards and the Smithsonian Institute, Washington, D. C. See par. 133 to par. 139a for definitions of abbreviations used in columns 8 and 9.

Liquid	Condition	Chemical Symbol	Spec. Gravity	Wt. in Lb. Per		Boiling Point ° F.	Fittings for Liquid End of Pump			
				Cu. In.	Gal.		Centrifugal	Displacement		
							Material	Material	Valves	Piston Pack.
1	2	3	4	5	6	7	8	9	10	11
Acid, Acetic	Conc.	$C_2H_4O_2$	1.055	0.0380	8.79	244.4	A. B.	A. B.	Disk	Bronze
Acetic............	Dil.	....	.....	......	.....	....	A. B.	A. B.	Disk	Bronze
Carbolic (Crude) .	...	C_6H_5OH	0.950 to 0.965	0.342 to 0.0348	7.91 to 8.04	182	A. I.	A. I.	Disk	Iron
Carbolic in Water.	...	CO_2	1.06	0.0382	8.83	....	P. F.	P. F.	Disk	......
Citric	...	$C_6H_8O_7$	1.54	0.0555	12.83	....	P. F.	P. F.	Disk	......
Cyanic..........	...	CONH	1.14	0.0411	9.52	....	A. I.	A. I.	Disk	Iron
Fatty(over130°F.)	...	$C_{10}H_{18}O_4$	.....	......	.....	....	A. B.	A. B.	Disk	Bronze
Hydrochloric.....	...	HCl	1.21	0.0436	10.08	....	Lead or A. B.	A. B.	Disk	Lead
Hydrocyanic.....	...	HCN	0.70	0.0252	5.83	....	A. I.	A. I.	Disk	Iron
Muriatic	...	....	1.21	0.0436	10.08	....	Lead or A. B.	A. B.	Disk	Lead
Mine Water	...	....	.....	......	.....	212	A. B.	A. B. or C. I. Wood or Lead Lined	Disk	Fibrous
Nitric...........	Conc.	HNO_3	1.52	0.0548	12.67	187	Lead or A. B.	A. B.	Disk	Bronze

140. PROPERTIES OF LIQUIDS AND FITTINGS FOR PUMPS—Continued

Liquid	Condition	Chemical Symbol	Spec. Gravity	Wt. in Lb. Per Cu. In	Wt. in Lb. Per Gal.	Boiling Point ° F.	Fittings for Liquid End of Pump: Centrifugal Material	Positive Displacement Material	Positive Displacement Valves	Positive Displacement Piston Pack.
1	2	3	4	5		7	8	9	10	11
Nitric...........	Dil.	....	.....	......	.....	....	A. I.	A. I.	Disk	Iron
Sulphuric........	Conc.	H_2SO_4	1.842	0.0664	15.35	640	A. I.	A. I.	Disk	Iron
Sulphuric..	Dil.	....	.....	......	.....	....	A .B.	A. B.	Disk	Bronze
Sulphuric	Fum	$H_2S_2O_7$	1.89	0.0681	15.76	....	A. I.	A. I.	Disk	Iron
Sulphuric 60° B...	...	....	1.71	0.0617	14.25	....	A. I.	A. I.	Disk	Iron
Sulphuric 40° B...	...	....	1.38	0.0497	11.50	....	Lead or A. B.	A. B.	Disk	Bronze
Sulphurous......	Conc.	H_2SO_3	.....	......	.....	....	....	....	Disk	......
Sulphurous......	Dil.	....	.....	......	.....	....	A. B.	A. B.	Disk	Bronze
Sulph. (Gaseous) .	...	....	.....	......	.....	....	F. B. F.	F. B. F.	Disk	Bronze
Alcohol, Grain (Ethyl).	...	C_2H_5OH	0.7939	0.0286	6.62	173	A. B.	A. B.	Disk	Hemp
Wood (Methyl) . .	...	CH_3OH	0.7965	0.0287	6.64	148	A. B.	A. B.	Disk	Hemp
Alkaline Liquid........	...	....	.....	......	.....	....	A. I.	A. I.	Disk	Iron
Alum................	...	$KAl(SO_4)_2$	1.64	0.0591	13.67	....	P. F.	P. F.	Disk	Hemp
Aluminum Sulphate....	...	$Al_2(SO_4)_3$	2.71	0.0977	22.59	....	F. B. F.	F. B. F.	Disk	Bronze
Ammonia—33.5.......	...	NH_3	0.60 0.62	0.0216 to 0.0224	5.01 to 5.16	—28.3	A. I.	A. I.	Disk	Iron
Wat. (Aq. Am.)	...	NH_3+H_2O	......	......	.....	....	A. I.	A. I.	Disk	Iron

140. PROPERTIES OF LIQUIDS AND FITTINGS FOR PUMPS—Continued

Liquid	Condition	Chemical Symbol	Spec. Gravity	Wt. in Lb. Per Cu. In.	Wt. in Lb. Per Gal.	Boiling Point ° F.	Fittings for Liquid End of Pump: Centrifugal Material	Positive Displacement Material	Positive Displacement Valves	Positive Displacement Piston Pack.
1	2	3	4	5	6	7	8	9	10	11
Ammonium Bicarbonate	...	NH_5CO_3	1.59	0.0574	13.24	....	P. F.	P. F.	Disk	Hemp
Chloride...	...	NH_4Cl	1.52	0.0548	12.67	....	A. I.	A. I.	Disk	Iron
Nitrate.....	...	NH_4NO_3	1.72	0.062	14.32	410	A. I.	A. I.	Disk	Iron
Sulphate...	...	$(NH_4)_2SO_4$	1.77	0.0638	14.72	....	A. I.	A. I.	Disk	Iron
Aniline Water.........	...	....	1.03	0.0371	8.57	....	A. I.	A. I.	Disk	Fibrous
Asphaltum............	Hot	....	0.98 to 1.4	0.0353 to 0.0504	8.16 to 11.65	....	....	A. I.	Ball	Iron
Barium Chloride.......	...	$BaCl_2$	3.86	0.139	32.1	....	....	P. F.	Disk	......
Nitrate........	...	$Ba(NO_3)_2$	3.24	0.1168	27.0	....	....	P. F.	Disk	......
Beer and Beer Wort....	...	....	.....	......	.....	....	A. B.	A. B. (F. B. F.)	Disk	Hemp
Beet Juice...........	...	....	.....	......	.....	....	F. B. F.	F. B. F.	Disk	Hemp
Benzene (Coal Tar Product...............	...	C_6H_6	0.88	0.0317	7.32	176	A. I.	A. I.	Disk	Iron
Benzine (Pet. Ether)...	...	CnH_2n+2	0.64 to 0.66	0.0232 to 0.0238	5.33 to 5.49	....	F. B. F.	F. B. F.	Disk	Rawhide
Bichloride of Mercury..	...	$HgCl_2$	5.3 to 5.5	0.192 to 0.1982	44.1 to 45.8	581	....	A. I.	Disk	Iron

140. PROPERTIES OF LIQUIDS AND FITTINGS FOR PUMPS—Continued

Liquid	Condition	Chemical Symbol	Spec. Gravity	Wt. in Lb. Per Cu. In.	Wt. in Lb. Per Gal.	Boiling Point ° F.	Fittings for Liquid End of Pump: Centrifugal Material	Positive Displacement Material	Positive Displacement Valves	Positive Displacement Piston Pack.
1	2	3	4	5	6	7	8	9	10	11
Bitterwater.	. . .	. . . .	. . . . .	. . . . . .	. . . . .	. . . .	A. B.	A. B.	Disk	Hemp
Brine, Calcium Chloride.	. . .	$CaCl_2$	. . . . .	. . . . . .	. . . . .	. . . .	All Iron or F. B. F.	F. B. F.	Disk	Hemp
Sodium.	. . .	. . . .	. . . . .	. . . . . .	. . . . .	. . . .	F. B. F.	F. B. F.	Disk	Hemp
Salt (3% Salt) . . .	. . .	NaCl	1.02	0.03675	8.49	. . . .	F. B. F.	F. B. F.	Disk	Hemp
Salt (over 3% Salt)	. . .	. . . .	1.02 to 1.20	0.03675 to 0.0432	8.49 to 10.00	. . . .	A. B.	A. B.	Disk	Hemp
Guncotton.	. . .	. . . .	. . . . .	. . . . . .	. . . . .	. . . .	. . . .	. . . .	. . . .	. . . . . .
Cachaza (Mud).	. . .	. . . .	. . . . .	. . . . . .	. . . . .	. . . .	. . . .	F. B. F.	Large Area	. . . . . .
Calcium Acid Sulphate. .	Conc.	$CaSO_4$	2.96	0.1068	24.6	. . . .	. . . .	A. B.	Disk	. . . . . .
Acid Sulphate. .	Dil.	. . . .	. . . . .	. . . . . .	. . . . .	. . . .	. . . .	F. B. F.	Disk	. . . . . .
Chlorate	. . .	$Ca(ClO_3)_2$	. . . . .	. . . . . .	. . . . .	. . . .	. . . .	A. B.	Disk	. . . . . .
Hypochlorite. . .	. . .	$Ca(OCl)_2$	. . . . .	. . . . . .	. . . . .	. . . .	. . . .	. . . .	. . . .	. . . . . .
Magn. Sod. Chl.	. . .	. . . .	. . . . .	. . . . . .	. . . . .	. . . .	F. B. F.	F. B. F.	Disk	Hemp
Carbonate of Soda.	. . .	Na_2CO_3	2.43 to 2.51	0.0876 to 0.0905	20.21 to 20.9	. . . .	. . . .	A. I.	Disk	Iron

140. PROPERTIES OF LIQUIDS AND FITTINGS FOR PUMPS—Continued

Liquid	Condition	Chemical Symbol	Spec. Gravity	Wt. in Lb. Per Cu. In.	Wt. in Lb. Per Gal.	Boiling Point ° F.	Fittings for Liquid End of Pump: Centrifugal Material	Positive Displacement Material	Positive Displacement Valves	Positive Displacement Piston Pack.
1	2	3	4	5	6	7	8	9	10	11
Carbon Tetrachloride	...	$C\,Cl_4$	1.58	0.057	13.15	....	....	P. F.	Disk	Hemp
Caustic Lye or Potash	...	K.OH	2.04	0.0735	17.0	....	....	A. I.	Disk	Iron
Manganese	...	$Mn(OH)_2$	.....	......	.....	....	....	A. I.	Disk	Iron
Soda	...	NaOH	2.13	0.0768	17.75	....	....	A. I.	Disk	Iron
Zinc Chloride	...	ZnClOH	.....	......	.....	....	....	F. B. F.	Disk	Hemp
Cellulose	...	....	1.27 to 1.61	0.0458 to 0.058	10.58 to 13.40	....	....	P. F.	Ball	Hemp
Chloride of Lime	...	$CaOCl_2$	.....	......	.....	....	....	....	Ball	......
of Zinc	...	$ZNCl_2$	2.91	1.05	24.20	....	....	A. I.	Disk	Iron
Chlorine and Water	...	....	.....	......	.....	....	....	....	Disk	......
Chloroform	...	$CHCl_3$	1.50	0.054	12.50	142	Lead	....	....	......
Copperas	...	$FeSO_4$	.....	......	.....	....	....	A. I.	Ball	Iron
Copper Nitrate	...	$Cu(NO_3)_2$	.....	......	.....	....	....	....	....	......
Sulphate	...	$CuSO_4$	3.52	0.1269	29.30	....	....	A. B.	Disk	Bronze
Creosote	...	....	0.93	0.0335	7.75	....	....	F. B. F.	Disk	Bronze
Cyanide Sodium	...	NaCN	.....	......	.....	....	....	A. I.	Disk	Iron

140. PROPERTIES OF LIQUIDS AND FITTINGS FOR PUMPS—Continued

Liquid	Condition	Chemical Symbol	Spec. Gravity	Wt. in Lb. Per Cu. In.	Wt. in Lb. Per Gal.	Boiling Point ° F.	Fittings for Liquid End of Pump: Centrifugal Material	Positive Displacement Material	Positive Displacement Valves	Positive Displacement Piston Pack.
1	2	3	4	5	6	7	8	9	10	11
Cyanide of Potassium	...	KCN	.....	......	.....	....	....	A. I.	Disk	Iron
Distillery Wort	...	....	.....	......	.....	....	A. B.	A. B.	Disk	Bronze
Dye Wood Liquor	...	....	.....	......	.....	....	....	F. B. F.	Disk	Hemp
Ethylene Chloride	...	$C_2H_4Cl_2$	1.28	0.0461	10.63	....	Lead	C. I. Lead Lined	Disk	Lead
Ferric Hydroxide	...	$Fe(OH)_3$	3.4 to	0.1224 to	28.3 to	....	....	A. B.	Disk	Bronze
			3.7	0.1335	30.8	....	....	....	Disk	Ebonite
Ferrous Chloride	...	$FeCl_2$	2.99	0.108	24.95					
Sulphate	...	$FeSO_4$	1.90	0.0685	15.82	....	....	A. I.	Ball	Iron
Gasolene	...	....	0.68 to	0.0245 to	5.66 to	158 to	....	F. B. F.	Disk	Rawhide
			0.75	0.027	6.25	194				
Glucose	...	....	1.56	0.0563	13.00	....	....	P. F.	Ball	Iron
Glue	Hot	....	.....	......	.....	....	....	F. B. F.	Ball	Bronze
Glycerine	...	$C_3H_5(OH)_3$	1.25	0.045	10.40	554	....	A. B.	Disk	Bronze
Grape Juice	...	....	.....	......	.....	....	....	A. B.	Disk	Hemp
Heptane	...	C_7H_{16}	0.68	0.0245	5.66	209	....	F. B. F.	Disk	Bronze
Hydrosulphite	...	HSO_2	.....	......	.....	....	Lead	C. I. Lead Lined	Disk	Lead
of Soda.	...	$Na_2S_2C_3$	.....	......	.....	....	Lead	C. I. Lead Lined	Disk	Lead

140. PROPERTIES OF LIQUIDS AND FITTINGS FOR PUMPS—Continued

Liquid	Condition	Chemical Symbol	Spec. Gravity	Wt. in Lb. Per Cu. In.	Wt. in Lb. Per Gal.	Boiling Point ° F.	Centrifugal Material	Positive Displacement Material	Positive Displacement Valves	Positive Displacement Piston Pack.
1	2	3	4	5	6	7	8	9	10	11
Lard (spec.gr. @ 60° F.)	Hot	....	0.92	0.0331	7.65	....	....	A. I.	Ball	Iron
Lead Nitrate..........	...	$Pb(NO_3)_2$	4.53	0.163	37.70	....	....	....	....	......
Lye..................	...	....	.....	......	.....	....	....	A. I.	Disk	Iron
Magnesium Chloride...	Hot	$MgCl_2$	2.18	0.0785	18.18	....	....	C. I. Lead Lined	Disk	Lead
Acid Sulph..	Conc.	....	.....	......	.....	....	....	A. B.	Disk	......
Acid Sulph..	Dil.	....	1.68	0.0605	14.00	....	....	F. B. F.	Disk	......
Oxychloride.	...	MgClOH	.....	......	.....	....	Lead	Lead or A. B.	Disk	......
Sulphate...	...	$MgSO_4$	2.65	0.0955	22.05	.....	....	A. I.	Disk	Iron
Magma (Thick Residue)	...	....	.....	......	.....	....	....	F. B. F.	Special	......
Marsh Gas(liq. @ 32° F.)	...	CH_4	0.55	0.0198	4.58	....	....	P. F.	Disk	......
Mash................	...	....	.....	......	.....	....	....	F. B. F.	Ball	......
Milk................	...	....	1.028 to 1.035	0.037 to 0.0373	8.55 to 8.62	....	....	A.B.	Disk	......
Milk of Lime..........	...	$Ca(OH)_2$	.....	......	.....	....	A. I.	A. I.	Ball	Iron
Molasses..............	...	....	.....	......	.....	....	....	F. B. F.	Ball	Bronze
Naphtha (Pet. Ether)..	...	C_nH_{2n+2}	0.665	0.024	5.54	....	....	F. B. F.	Disk	......

140. PROPERTIES OF LIQUIDS AND FITTINGS FOR PUMPS—Continued

Liquid	Condition	Chemical Symbol	Spec. Gravity	Wt. in Lb. Per Cu. In.	Wt. in Lb. Per Gal.	Boiling Point °F.	Fittings for Liquid End of Pump: Centrifugal Material	Fittings for Liquid End of Pump: Positive Displacement Material	Positive Displacement Valves	Positive Displacement Piston Pack.
1	2	3	4	5	6	7	8	9	10	11
Nickel Chloride........	...	$NiCl_2$	2.56	0.0923	21.30	....	....	A. B.	Disk	Bronze
Sulphate........	...	$NiSO_4$	3.42	0.123	28.45	....	....	A. B.	Disk	Bronze
Oil Crude Asphal. Base CrudeParaffineBase Light Lubricating Heavy Lubricating Mineral or Veg...	Hot or Cold	....	.....	......	.....	....	P. B. or B. F. A. I. Varies with Service	P. B. or B. F. A. I. Varies with Service	Varies with Temp. & Press. Disk Ball or Wing	Varies with Temp. Iron or Bronze Three Ring
Creosote..........	...	....	1.04 to 1.10	0.0375 to 0.396	8.67 to 9.16	....	....	F. B. F. or A. I.	Disk	Bronze
Cotton Seed........	...	....	.....	......	.....	....	....	P. F.	Disk	Hemp
Coal Tar..........	...	....	.....	......	.....	....	....	....	....	......
Cocoanut..........	...	....	.....	......	.....	....	....	A. I.	Disk	......
Fuel..............	...	....	.....	......	.....	....	....	P. F. or B. F.	Disk	Iron
Gas..............	...	....	.....	......	.....	....	....	A. I.	Disk	Iron
Kerosene..........	...	....	.....	......	.....	....	P. F.	P. F.	Disk	Iron

140. PROPERTIES OF LIQUIDS AND FITTINGS FOR PUMPS—Continued

Liquid	Condition	Chemical Symbol	Spec. Gravity	Wt. in Lb. Per		Boiling Point ° F.	Fittings for Liquid End of Pump			
							Centrifugal	Positive Displacement		
				Cu. In.	Gal.		Material	Material	Valves	Piston Pack.
1	2	3	4	5	6	7	8	9	10	11
Linseed	...	....	0.94	0.0339	7.84	....	....	B. F.	Disk	Hemp
Rapeseed	...	....	0.92	0.0331	7.66	....	A. I.	A. I.	Disk	Iron
Turpentine	...	....	0.87	0.0314	7.25	....	A. I.	A. I.	Disk	Iron
Wash	...	....	.....	......	.....	....	....	A. I.	Disk	Iron
Paraffine	Hot	....	.....	......	.....	662 to 806	....	F. B. F.	Disk	Bronze
Peroxide of Hydrogen	...	H_2O_2	.....	......	.....	176	A. B.	A. B.	Disk	Bronze
Petroleum	...	....	0.88 to 0.92	0.0317 to 0.0331	7.33 to 7.66	....	See Oils	See Oils	....	......
Potash	...	K_2CO_3	.....	......	.....	....	....	A. I.	Disk	Iron
Sulphide	...	K_2S	.....	......	.....	....	....	A. I.	Disk	Iron
Potassium Alum	...	$Al_2K_2(SO_4)_4$	.....	......	.....	....	....	A. I.	Disk	Iron
Carbonate	...	K_2CO_3	2.33	0.084	19.40	....	....	A. I.	Disk	Iron
Chloride	...	KCl	1.99	0.0717	16.57	....	....	A. B.	Disk	Bronze
Cyanide	...	KCN	1.52	0.0548	12.66	....	....	A. I.	Disk	Iron
Nitrate	...	KNO_3	2.11	0.076	17.55	....	....	A. B.	Disk	Bronze
Sulphate	...	K_2SO_4	2.66	0.096	22.18	....	....	A. B.	Disk	Bronze

140. PROPERTIES OF LIQUIDS AND FITTINGS FOR PUMPS—Continued

Liquid	Condition	Chemical Symbol	Spec. Gravity	Wt. in Lb. Per		Boiling Point ° F.	Fittings for Liquid End of Pump			
							Centrifugal	Positive Displacement		
				Cu. In.	Gal.		Material	Material	Valves	Piston Pack.
1	2	3	4	5	6	7	8	9	10	11
Sal Ammoniac	...	NH_4Cl	1.52	0.0548	12.65	...	...	A. I.	Disk	Iron
Sea Water	...	...	1.03	0.0371	8.57	...	F. B. F.	F. B. F.	Disk	Hemp
Sewage	...	...	...	...	...	...	F. B. F.	F. B. F.	Special	Hemp
Size	...	...	...	...	...	...	...	A. B.	Disk	Bronze
Slop, Brewery	...	...	...	...	...	...	...	F. B. F.	Special	Hemp
Soap	...	...	...	...	...	...	...	A. I.	Ball	Iron
Soda	...	NaOH	...	...	...	...	...	A. I.	Disk	Iron
Sodium Bicarbonate	...	$NaHCO_3$	...	...	...	...	...	A. I.	Disk	Iron
Carbonate	...	Na_2CO_3	2.43 to 2.51	0.0876 to 0.0904	20.20 to 20.90	...	...	A. I.	Disk	Iron
Chloride	...	NaCl	2.16	0.0778	18.0	8114	...	A. B.	Disk	Bronze
Hydroxide	...	NaOH	2.13	0.0767	17.70	...	...	...	...	...
Hyposulphite	...	$NaHSO_2$	...	...	...	...	...	A. I.	...	...
Nitrate	...	$NaNO_3$	2.27	0.0817	18.90	716	...	...	...	...
Sulphate	...	Na_2SO_4	2.67	0.0962	22.20	...	...	A. I.	Disk	...
Starch	...	...	...	...	...	...	...	F. B. F.	...	...
Strontium Nitrate	...	$Sr(NO_3)_2$	2.98	0.1072	24.80	...	...	A. B.	Disk	Bronze

140. PROPERTIES OF LIQUIDS AND FITTINGS FOR PUMPS—Continued

Liquid	Condition	Chemical Symbol	Spec. Gravity	Wt. in Lb. Per		Boiling Point ° F.	Fittings for Liquid End of Pump			
				Cu. In.	Gal.		Centrifugal	Positive Displacement		
							Material	Material	Valves	Piston Pack.
1	2	3	4	5	6	7	8	9	10	11
Sugar	...	....	.....	......	.....	....	....	A. B.	....	......
Sulphate of Lime	...	$CaSO_4$	.....	......	.....	....	....	A. B.	....	......
Sulphide of Hydrogen	...	H_2S	.....	......	.....	....	....	P. F.	....	......
of Sodium	Hot	Na_2S	1.86	0.067	15.47	....	....	A. I.	Disk	Iron
of Sodium	Cold	....	.....	......	.....	....	....	P. F.	Disk	Hemp
Sulphur Dioxide	...	SO_2	.....	......	.....	....	....	A. B.	Disk	Bronze
Syrup	...	....	.....	......	.....	....	....	A. B.	....	......
Tan Liquor	...	....	.....	......	.....	....	....	A. B.	Disk	Bronze
Tar	Hot	....	.....	......	.....	....	....	A. I.	Ball	Iron
Tar and Am. in Water	...	....	.....	......	....	...	....	A. I.	Disk	Iron
Varnish	...	....	.....	......	.....	....	....	F. B. F.	Disk	Bronze
Vinegar	...	....	.....	......	.....	...	....	A. B.	Disk	Bronze
Vitriol, Blue	...	$CuSO_4$	.....	......	.....	....	....	A. B.	Disk	Bronze
Green	...	$FeSO_4$	.....	......	.....	....	....	A. I.	Ball	Iron
Water, Fresh	...	....	1.00	0.0360	8.335	212	B. F.	B. F.	Disk	Hemp

140. PROPERTIES OF LIQUIDS AND FITTINGS FOR PUMPS—Concluded

Liquid	Condition	Chemical Symbol	Spec. Gravity	Wt. in Lb. Per Cu. In.	Wt. in Lb. Per Gal.	Boiling Point °F.	Fittings for Liquid End of Pump: Centrifugal Material	Positive Displacement Material	Positive Displacement Valves	Positive Displacement Piston Pack.
1	2	3	4	5	6	7	8	9	10	11
Salt (Sea)	Cold	....	1.03	0.0371	8.57	....	F. B. F.	F. B. F.	Disk	Hemp
Salt (Sea)	Hot	....	1.02	0.03675	8.49	....	F. B. F.	F. B. F.	Disk	Hemp
Distilled	...	....	1.00	0.0360	8.335	212	F. B. F.	F. B. F.	Disk	Hemp
Dish	...	....	.....	......	.....	....	P. F.	P. F.	Disk	Hemp
Lime	...	$Ca(OH)_2$	.....	......	.....	....	A. I.	A. I.	Disk	Iron
Acid Mine	...	....	.....	......	.....	....	A. B.	A.B. C. I. Wood or Lead Lined	Disk	......
Sweet	...	....	.....	......	.....	....	F. B. F.	F. B. F.	Disk	Hemp
Sulphur	...	....	.....	......	.....	....	F. B. F.	F. B. F.	Disk	Hemp
Wood Pulp	...	....	.....	......	.....	....	F. B. F.	F. B. F.	Ball	Hemp
Yeast	...	....	.....	......	.....	....	A. B.	A. B.	Ball	Bronze
Zinc Nitrate	...	$Zn(NO_3)_2$	2.06	0.0742	17.10	268	....	A. B.	Disk	Bronze
Sulphate	...	$ZnSO_4$	3.62	0.1302	30.10	....	....	A. B.	Disk	Bronze

SECTION II

Centrifugal Pumps

(Figures refer to paragraph numbers)

Types and design problems, 1-43; Open-impeller Class C Pump, 44-48; Sump pump, Class C, 49-50; Volute pumps, double suction Class OS, 51-65; Multi-stage pumps, Class SD, 66-79; Multi-stage pumps, Class JDS, 80-99; Ball-bearing Volute pumps for general service, Types H, L, M, R, S and U, 100-127; Special-service pumps, 154-158; Drainage pumps, 158-161; Irrigation pumps 162-165; Underwriter fire pumps, 166-170; Open-impeller pumps, Class OS and BS ,173-176; Pulp pumps, Class P, 177-178; Special service volute pumps, 179-183; High-pressure pumps, 184; Vertical pumps, 185.

SECTION II

CENTRIFUGAL PUMPS

1. The centrifugal pump consists of two major parts: A fixed casing with stuffing boxes, bearings, etc., and a rotating member consisting of the shaft on which is keyed one or more impellers depending on the type of pump. The impeller is sometimes called the runner or wheel.

2. In the **early** stages of its **development** the centrifugal was a single-stage pump used only for large volumes of water at low heads. The centrifugal pump was more suited to this work than the positive-displacement pump as it was lighter in weight, less expensive and occupied less floor space.

3. Manufacturers and engineers, however, were not content with the limited application of the centrifugal pump and were constantly experimenting with new designs and types in an effort to enlarge its field of usefulness. The results of these experiments led to many improvements and refinements in the single-stage pump and also led to the development of the multi-stage turbine pump for high heads. During the period of this development the field of application of the centrifugal was gradually increased and today there is a centrifugal pump available for practically every service and for all capacities and heads up to 1500 ft.

4. **Types.**—Centrifugal pumps are divided into two general classes or styles: The single-stage volute pump for low and medium heads and the multi-stage pump for high heads.

5. The **volute pump** has a plain spiral or volute casing and is subdivided as to impeller design into:

$$\text{Single-Suction}\begin{cases}\text{Open impeller}\\ \text{Closed impeller}\end{cases}$$

$$\text{Double-Suction}\begin{cases}\text{Open impeller}\\ \text{Closed impeller}\end{cases}$$

6. The double-suction type with its hydraulic balance is the preferred type of volute pump.

7. **Multi-stage pumps** (excepting special cases) have a circular casing in which diffusion vanes are sometimes used; they are then called turbine pumps. The multi-stage pump may be subdivided as to impeller design into single suction and double suction.

8. The single-stage turbine pump has been replaced by the double-suction volute pump which will in nearly all cases meet the service conditions for which a single-stage turbine could be used.

9. The multi-stage pump employs two or more impellers in series operation to multiply the head. To illustrate this, let us take a four-stage pump operating with 440 ft. discharge head. The suction or No. 1 impeller would raise the water from suction to 110 ft. at which head it would be discharged into No. 2 impeller which would add another 110 ft. to the head and discharge the water into No. 3 impeller at 220 ft. head, and No. 3 would add another 110 ft. to the head and deliver the water into No. 4 impeller which will add the final 110 ft. to the head and will deliver the water to the discharge of the pump at the required 440 ft. head.

10. **Pump Sizes.**—Any or all classifications as to type may apply to one pump but a universal designation as to its size is made by stating the diameter of the discharge opening. However, the size of the discharge opening is not a direct measure of the capacity of a centrifugal pump. The capacity varies with the head pumped against, the area and form of the suction passages of the pump and the suction conditions external to the pump.

11. The **terms** universally used in centrifugal-pump practice are: Head, capacity, brake-horsepower, and efficiency, all of which are explained in Section I.

12. The **basic law** of centrifugal-pump operation is centrifugal force; which is a function of mass and the square of velocity.

13. The **impeller** is the heart of a centrifugal pump. It is the agent employed to obtain the rotation of the mass of liquid, and the peripheral speed of its vane tips when squared gives the measure of the head produced, which is velocity head.

14. **Impeller Design.**—The greatest care and skill are employed in the designing, molding, machining and finishing of the impellers used in Worthington Centrifugal Pumps. The impellers are

designed with passages of the proper area to give a smooth, even flow to the liquid. The careful hand-finish gives a smooth surface which reduces skin friction to the minimum. Proper proportions and vane shapes prevent contractions and shocks. All Worthington impellers (excepting special cases) are made from a special composition bronze which is unquestionably the best metal to withstand the action of liquids and at the same time retain its original shape and smoothness. Iron impellers are necessary in some cases but their use is to be avoided as far as possible. Iron always corrodes and as a result the smooth finish is soon lost, causing a marked decrease in the efficiency of the pump. For pumps handling large quantities of liquids against low heads, with a low number of revolutions, per minute the iron impeller is, however, quite suitable.

15. Size and type of pump are minor factors, with speed, head and capacity as the major factors of impeller design. The liquid to be pumped determines whether the open or closed impeller will be used.

16. The **form of the vane** divides the impeller into its distinctive type such as (1) straight vane, (2) Francis mixed flow and (3) screw, or axial-flow, propeller type. The type of impeller can also be expressed as the ratio between inlet and outlet diameters or as the ratio of impeller diameter to width of the impeller at the discharge.

17. To better understand the **influence of speed** on impeller design, let us consider the motor speeds for 60-cycle motors which are the most used for driving centrifugal pumps. The standard speeds are 900, 1200, 1800 r.p.m. and 3600.* On account of weight and cost it is desirable to use the high-speed motors whenever possible. All Worthington motor-driven centrifugal pumps up to and including the 12-in. can be driven by 1800-r.p.m. motors, except where the pumping head is relatively low. The pump speed is always selected so as to result in such impeller dimensions as will produce maximum efficiencies. For pumps larger than 12-in. there are very few instances where 1800 r.p.m. will comply with this requirement and it will be found that 1200 r.p.m. or even lower speeds are a better selection. With a fixed speed in revolutions per

*See page 49 on Ball-bearing Pumps

minute, the head remaining the same, the specific speed (see par. 37) will increase with the size of the pump since the capacity in cubic feet per second (Q) also increases with the size of the pump. Up to and including 10-in. pumps, the straight-vane impeller, Fig. 1, will meet the requirements of good, efficient design for most conditions of service. The higher specific speeds for pumps of 12 in. and above require an impeller having a low pick-up loss at the suction entrance.

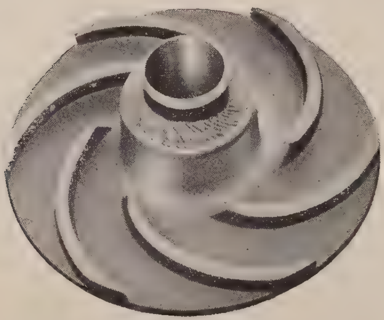


FIG. 1. Impeller, straight-vane type.

18. The **Francis mixed-flow impeller,** Fig. 2, having the vanes extended across the suction eye of the impeller, meet the requirement of low pick-up loss at high rotative speeds. The entrance angles of the vanes change with radii and section, giving a uniform absolute velocity across the vane section, which results in a full capacity flow without cavitation or undue shock, in high efficiency and a full characteristic. A sudden break in the head characteristic indicates the use of a poorly designed impeller with an interference or stopping of accelerated flow resulting from excessive eddy currents and excessive pick-up losses.

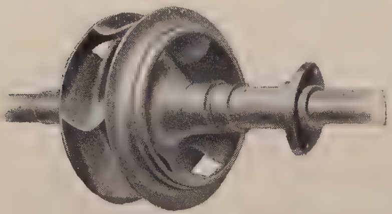


FIG. 2. Impeller with Francis mixed-flow vanes.

19. The **Francis extreme** type of impeller, Fig. 3, is limited to operating heads of about 45 ft. The highest-speed impeller of this type is obtained when the discharge diameter is equal to or slightly less than the inlet diameter. The pick-up speed of the liquid

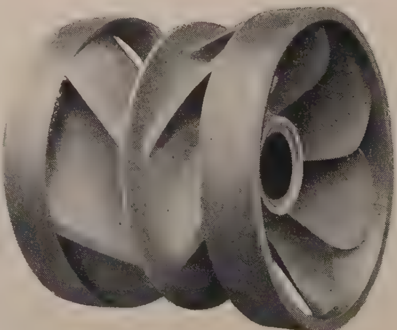


FIG. 3. Impeller with Francis extreme vanes.

at the inlet diameter is equal to the peripheral speed of an impeller of this type. This pick-up or inlet speed has a maximum beyond which it is unsafe to operate a pump, due to erosion and the possibility of cavitation.

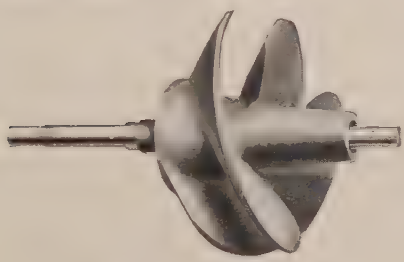


Fig. 4. Impeller with screw vanes.

20. For large capacities and heads ranging from 5 to 45 ft., such as used in drainage and irrigation work, an impeller of the **screw type,** Fig. 4, is used to advantage. While we have the same limit of pick-up speed, this limit with a screw impeller on a 5-ft. head allows of a specific speed about three times that of the extreme Francis type. The mean delivery diameter of the screw impeller is much less than the inlet diameter and the flow through the impeller is nearly axial. The screw impeller has the highest specific speed of any impeller as yet produced.

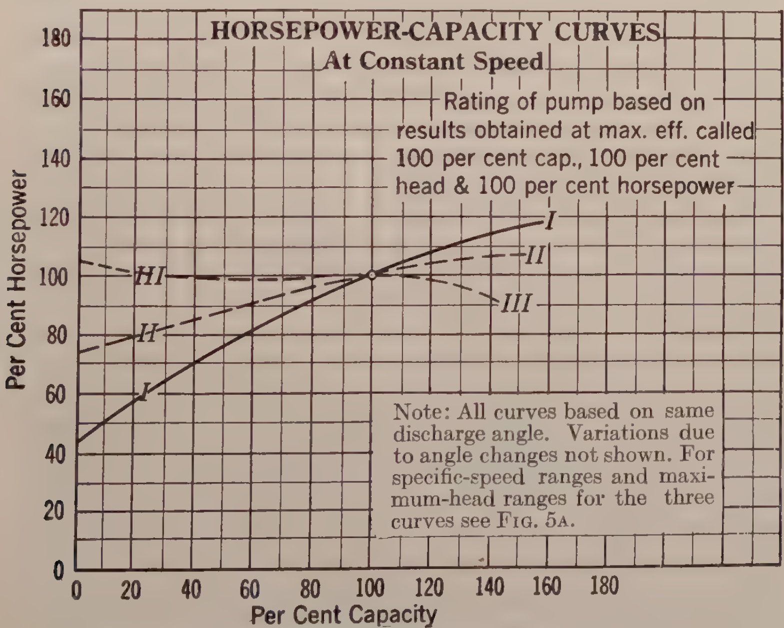


Fig. 5. Effect of specific speed upon horsepower-capacity curves.

21. A comparison of the effects on the characteristic curves by changes in the **specific speed** of impellers is shown by Figs. 5 and 5A. The head and horsepower curves show that as the specific speeds increase, and the discharge diameter of the impeller approaches the inlet diameter, the greater the increase in head and horsepower demand at shut-off or zero capacity head. This has an important bearing on the use of synchronous motors for driving centrifugal pumps. When the pump is primed with the discharge valve closed

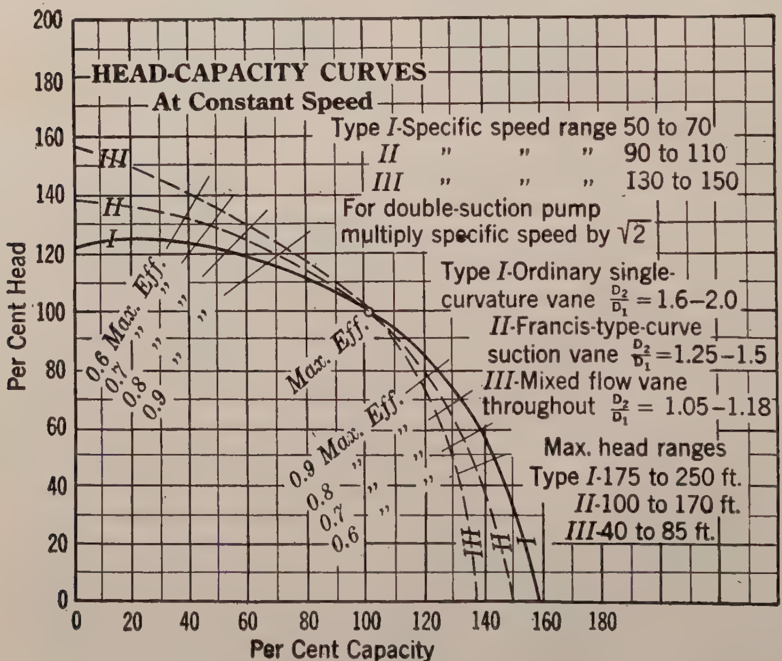


FIG. 5A. Effect of specific speed upon head-capacity curves.

and brought up to speed it is consuming the power shown at the shut-off point. If the shut-off power is equal to (or greater than) the normal horsepower rating of a synchronous motor it is impossible to get such a motor to pull into step. Therefore, for synchronous-motor drives the lower specific-speed impellers are used.

22. Where it is desirable to use a synchronous motor for power-factor correction and there is some doubt about the starting load,

a Fynn-Weichsel motor may be used. The Fynn-Weichsel motor will start and pull into synchronism a load about 50 per cent greater than full load. It also will run as an induction motor on loads between 150 per cent and 300 per cent full load, and whenever the overload drops to less than 150 per cent, it will pull itself back into synchronism.

25. **Casing Design.**—If, after leaving the impeller, the rotating mass of liquid be restrained by a casing, it will generate pressure due to such restraint. In a well designed and well proportioned volute casing this change from velocity head to pressure head is brought about gradually and without shock, giving the centrifugal pump the much desired smooth discharge flow without fluctuation in pressure.

26. Casing and impeller design are closely related to each other. A **coefficient of velocity** throughout the passages of the casing and the impeller, which coefficient is determined from the square root of the head, indicates a constant for the design of impellers proportioned to each other. In order to keep the absolute velocities from reaching impossible values on high heads, it has been found necessary in practice to vary this coefficient with the magnitude of the head. For example, if the coefficient of a velocity is "K" and represents a velocity of 12 ft. per sec. on head of 100 ft., this same coefficient would represent a velocity of 17 ft. per sec. on a head of 200 ft. The eddies set up in the suction nozzle of a casing by this velocity of 17 ft. per sec. are so much out of proportion to those existing when the velocity is 12 ft. per sec. that the laws of homologous design will not apply unless some provision is made for taking care of the higher values. The application of this principle is illustrated in the design of the double-suction Class OS volute pump. The small sizes, 2 to 6-in., Fig. 19, have elbow type suction nozzles which give perfectly satisfactory service at the capacities and heads for which these pumps are used. The larger sizes, 8-in. and upward, Fig. 20, have the volute-type of suction nozzles. This volute-type suction nozzle is also used on all double-suction pumps designed for heads of 100 ft. and upward.

27. The ordinary **double-suction pump** involves the division of the water around the two sides of the casing to the entrance of the impeller, and the recombining of these two streams into the volute

casing. This division and recombining causes a loss which is not perceptible at heads of 20 ft. and up. In low-head range of 5 ft. to 20 ft. the loss is quite perceptible. Under these conditions the casing is designed with two separate suction elbows, to reduce these losses and to obtain greater capacities from the pump at these low heads. This design is applied to pumps for drainage and irrigation service. (See Fig. 36.)

Fig. 6. Diffusion ring.

28. Due to the circular form of the casing of the multi-stage turbine the conversion of velocity head to pressure head is brought about by inserting a **diffusion ring,** Fig. 6, between the impeller and the casing. The diffusion rings are designed with vanes arranged so as to give a gradual increase in the area of the liquid passage to reduce the velocity of the liquid leaving the impeller, and to convert this velocity head efficiently into pressure head. These passages are in effect a number of inverted Venturi throats surrounding the impeller.

29. Head.—A heavy liquid having greater mass generates greater pressure as compared to water but does not generate a greater head in feet of the liquid pumped. There are other elements that enter into the problems of head generation which are of the nature of head produced by velocity changes. These elements also occur in the square relation of velocity to head (see par. 33) so that by applying the proper coefficient to the peripheral speed of an impeller, the head it will generate is definitely known.

30. Capacity and Speed.—The quantity of liquid delivered depends on the area of the passages through the impeller. As quantity is a function of area and velocity, it is evident that with a given area in an impeller, the vane velocities will increase with any increase in the quantity of liquid delivered by that impeller. This velocity is a part of the energy-producing head. As the quantity increases a point will be reached where all the energy that would otherwise appear as head goes into velocity and a zero net head would result. It is, therefore, apparent that the relations between the peripheral

speed of the impeller, the impeller area, and the capacity are always definite and always tied together. This relation showing the variation in capacity with corresponding head at a fixed speed, when expressed graphically forms the characteristic curve of the pump.

31. Characteristic Curves.—By the law of homologous design, pumps of the same type and similar design of impellers have characteristic curves of the same shape, and, when these values are expressed in percentage relation, give the same curve.

Characteristic curves are based on average test results of a given type of impeller and are applicable to any impeller of that type with the reservation that being an average, they cannot be expected to be guaranteed throughout their entire range for any particular case.

32. These curves indicate the changing relationship between head, capacity, speed, efficiency and brake-horsepower, corresponding to variations of one or more of these factors, and are drawn up for the operating range within which changes can be predicted with a satisfactory degree of accuracy.

Fig. 7 is a typical 100 per cent characteristic curve and is thus explained:

100 per cent speed—normal speed.

100 per cent head—normal designed head where maximum efficiency occurs and 100 per cent speed.

100 per cent hp.—horsepower at normal head and capacity or 100 per cent of rated power at normal speed.

100 per cent efficiency—normal maximum efficiency or 100 per cent of the best efficiency obtained, and applies throughout a range of speed on parabolic coordinates.

33. An inspection of this curve at a given point, say the 100 per cent point, shows that with a given impeller the **output** varies with the **speed** in the following ratios:

Head	varies	as the	second	power	of	the	speed	variation
Capacity	"	"	first	"	"	"	"	"
Brake-horsepower	"	"	third	"	"	"	"	"

In order to make these curves more conveniently applicable to a

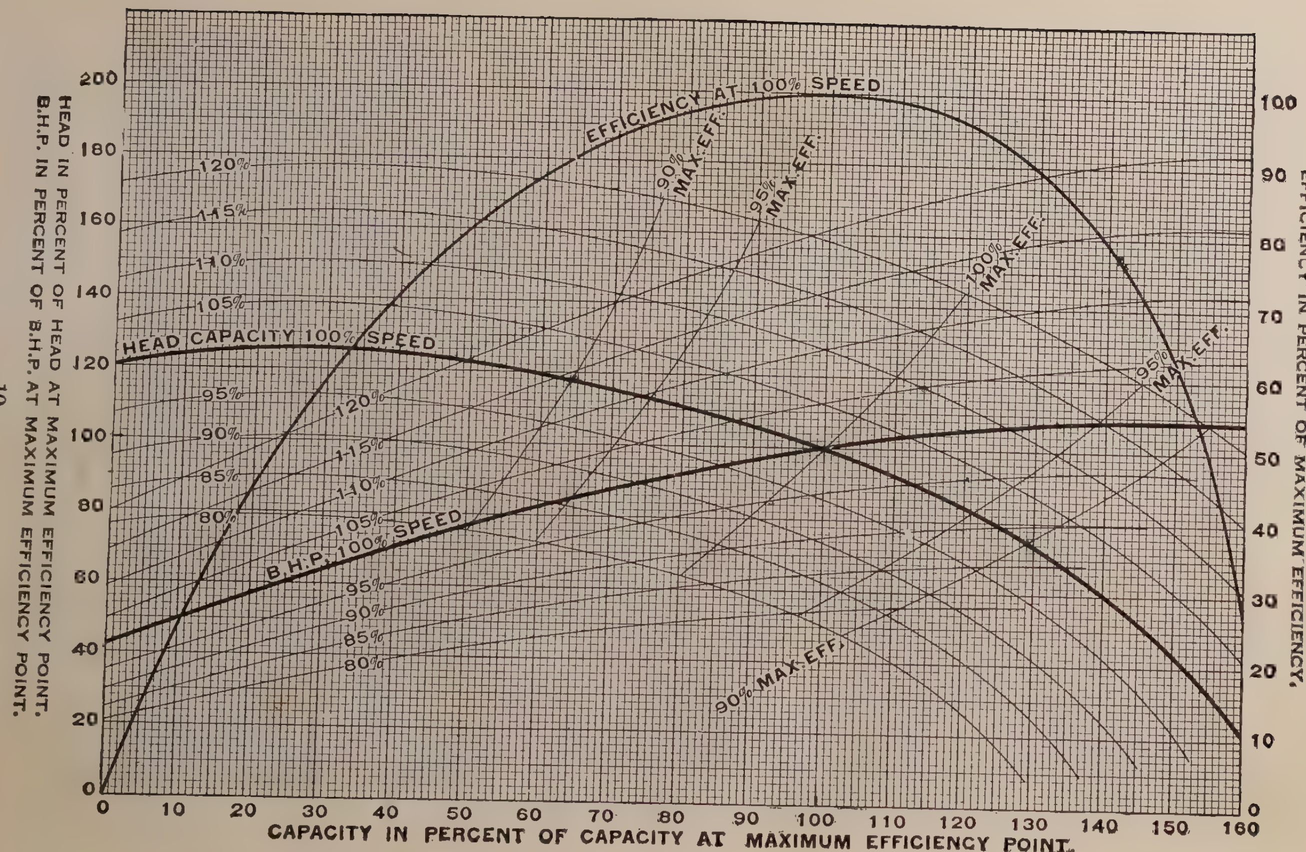


Fig. 7.

Typical 100 per cent Characteristic Curve.

particular case, the ordinates and abscissae are given in terms of percentages of head, capacity, brake-horsepower and efficiency for the operating point at which the pump will be most efficient. After this point has been determined from a previous series of tests, any other point of operation can be closely determined by simple proportion, as explained further on in this discussion.

In discussing the use of these curves the following notation will be used: h = head e = efficiency n = speed
c = capacity p = brake-horsepower

$\%h$ = head in per cent of head at maximum efficiency point read at left-hand margin of curve.

$\%c$ = capacity in per cent of capacity at maximum efficiency point read at bottom margin of curve.

$\%e$ = efficiency in per cent of efficiency at maximum efficiency point read at right-hand margin of curve.

$\%p$ = brake-horsepower in per cent of brake-horsepower at maximum efficiency point read at left-hand margin of curve.

$\%n$ = speed in per cent of speed at original condition of operation read on head-capacity and brake-horsepower-capacity curves.

Subscript 1 = original condition of operation.

Subscript 2 = new condition of operation.

It is first necessary to mark the position of the original (or normal) condition of operation h_1 c_1 e_1 as shown by Fig. 7, in order to have a starting point from which to base the calculations for any other condition of operation. This point is to be indicated on the head-capacity curve drawn up for 100 per cent or normal operating speed.

In making calculations for new points of operation, after this point has been indicated on the curve, the corresponding percentages as $\%h_1$, $\%c_1$, $\%e_1$, and $\%p_1$, are next to be read off at the proper margin and the procedure will then be as indicated in the examples given in Fig. 8 to 15 inclusive, the general rule for determining any new factor being shown by the relation

$$\frac{h_2}{h_1} = \frac{\%h_2}{\%h_1}$$

34. For any operating point, the required brake-horsepower will be given as $\%p$ at the intersection of the $\%c$ abscissa with the horsepower curve drawn up for the speed at which the head and capacity are to be obtained

35. For any operating point at normal or 100 per cent speed the corresponding efficiency will be given at the intersection of the $\%c$ abscissa with the efficiency curve drawn up at 100 per cent speed. For any other speed, the rules are given in Fig. 13 to Fig. 15 inclusive for the use of these curves at variable speed.

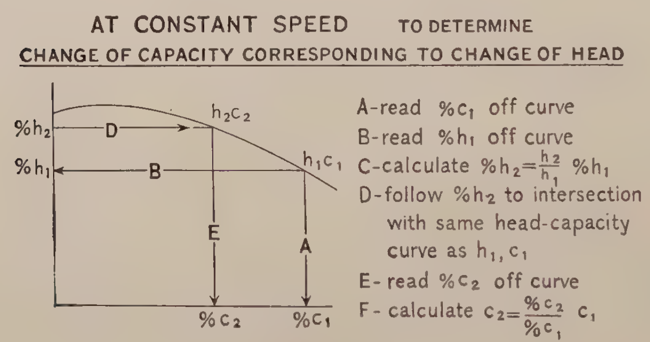


FIG. 8.

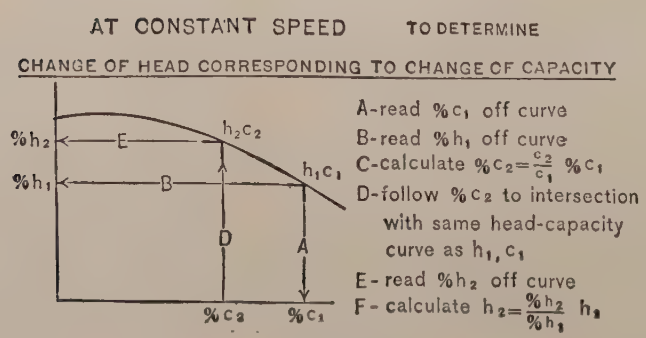


FIG. 9.

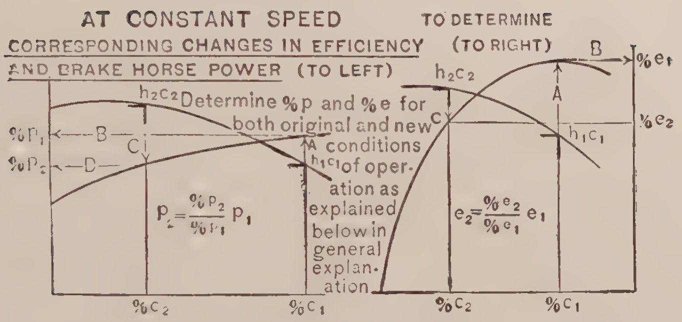


Fig. 10.

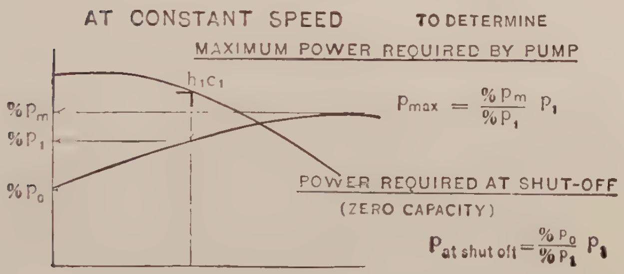


Fig. 11.

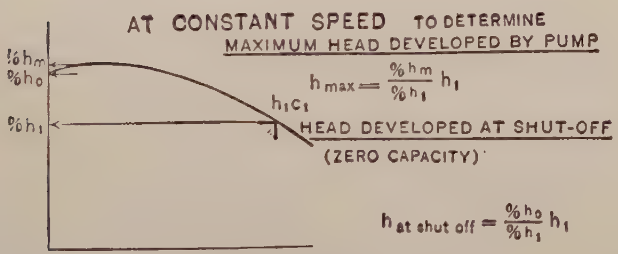


Fig. 12.

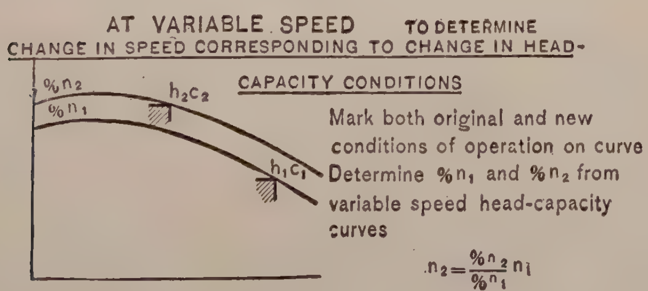


Fig. 13.

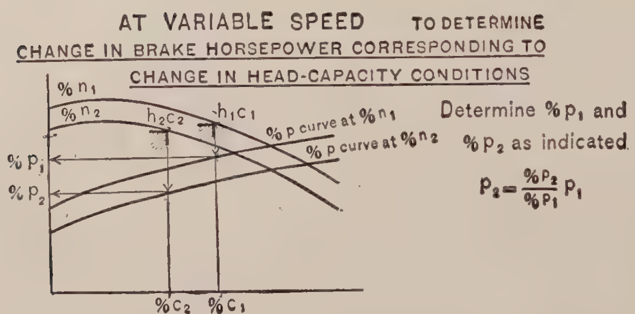


FIG. 14.

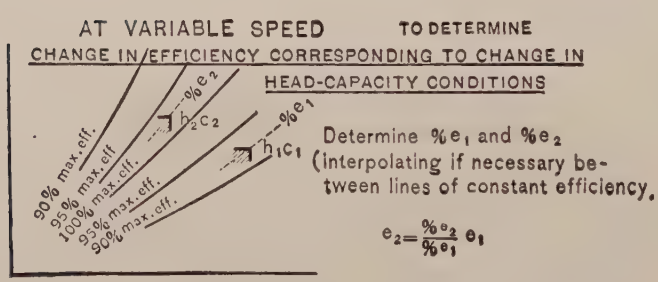


FIG. 15.

36. Design of Centrifugal Pumps.—It has been demonstrated that from an hydraulic viewpoint the only correct design for a pump is one that is based on the results of experiment and test. The Worthington Centrifugal Pump is the product of years of research and tests wherein the changes in effect resulting from changes in the form and shape of casings, impellers, suction passages and other details have been carefully and intelligently analyzed and applied to new designs. During this research work, which was made possible by the Worthington testing laboratory with facilities for a full range of capacity and pressure tests with various drives, it was found that the results obtained by testing a number of impellers of different types in a model casing were insufficient to meet all the requirements of good design. In order to obtain the

required data, casings of different sizes and types and of different forms and shapes of both casing and suction passages were tested with various types of impellers. These experiments proved that the change in effect resulting from changes in casing and impeller design varied with different sizes of pumps, and that by testing a number of sizes the exact relation of a given size and type of casing to a given type of impeller could be definitely fixed. This research work also enabled Worthington to establish definitely the relation between impeller proportion and specific speed.

37. Specific speed is a relation between capacity, speed and head and has a definite bearing on the proportions of an impeller to produce the designed results. It may be expressed by the formula

$$N_s = \frac{NQ^{\frac{1}{2}}}{H^{\frac{3}{4}}} \text{ or } N_s = \frac{r.p.m.}{\sqrt{\mathrm{H}}} \times \sqrt{\frac{Q}{\sqrt{\mathrm{H}}}}$$

N_s = specific speed
N = r.p.m.
Q = capacity in cu. ft. per sec.
H = head in feet (water)

Note.—This formula is not to be used for computing the specific speed of hydraulic turbines.

38. In general N and Q could be varied for a given head for the same value of N_s but impeller proportions are so tied up with these values that it makes the subject difficult to express in the usual forms of engineering formula. It is only through years of experience that the designer is enabled to tie up impeller proportions definitely with specific speed. For this reason many manufacturers of centrifugal pumps have confined their efforts to the more simple and commonly known types of impellers.

39. It is not alone years of experience in building pumps, but years of experiments, and the careful study of the performance of pumps that has made the Worthington Centrifugal Pump the standard with which others are compared.

40. Standard Pumps.—Another outstanding development resulting from research work is the Worthington Standard Centrifugal Pump. The data accumulated have not only enabled Worthington to take full advantage of changes in effect resulting from changes in impeller design but to fix definite limits of service for each

type and size of standard pump, thereby assuring the purchaser of always getting "the proper pump for the service."

41. The term standard is used here to designate a pump that will meet the greatest service range with a given casing and at the same time be designed so that practically all parts may be built and carried in stock.

42. Worthington Standard Centrifugal Pumps are now available in the following types for heads up to 925 ft. and for belt, motor or steam-turbine drives.

	Type	Class	Service	Head in Ft.
Single-Stage.........	Single-Suction	R S C	General Contractors Small irrigation	10– 80
Volute..............	Double-Suction	H L M OS	General Water Works Booster	120 10–120 100–200
Two-and Four-Stage Volutes...........	Single-Suction	SD U	Fire Pumps Elevator General	175–600
Multi-Stage Turbine..	Single- and Double-Suction	JDS	Boiler-Feed and High pressure	350–925

43. With the exception of the single-suction Class C volutes all of the above types can be modified to meet special conditions of service. The Class OS can be made with open impellers for pumping pulp and similar liquids with solids in suspension, as will be explained later under pumps for special service.

44. Class C Volutes. For low-head service (80 ft. and under) where refinements are unnecessary and are not wanted, such as contracting and intermittent service, small irrigation or drainage units, the Worthington Class C Pump has been developed in sizes from 1 to 8 in. inclusive.

45. This is an overhung open-impeller type pump. The **construction** of the 1 and 1½-in. is shown by Fig. 17A. Sizes 1-in. and 1½-in. can be furnished for belt drive or direct-connected to electric

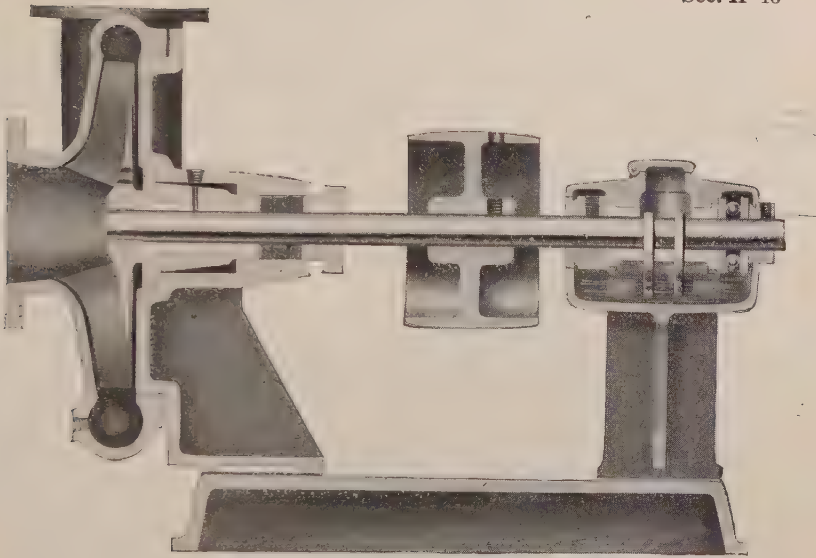


Fig. 16. Cross section of Worthington Class C Over-hung Open-impeller Centrifugal Pump.

motors. The motor-driven pumps make ideal units for small capacities at low and medium heads. The small amount of end thrust is taken by the hub of the pulley or coupling. The construction of sizes 2 to 8-in. is shown by Fig. 16 and 17. These sizes are furnished for belt drive only.

46. The **casings** are of the solid type and with the head form the shrouding for the impeller. Each size pump is furnished with one

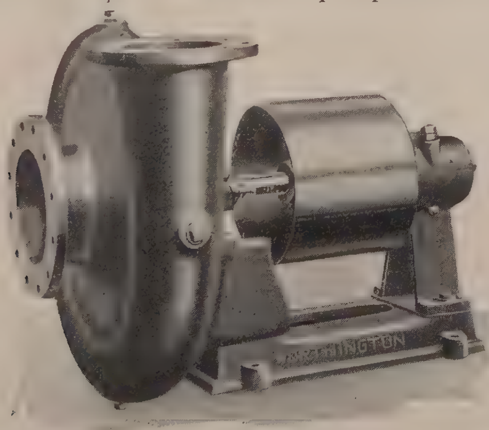


Fig. 17. Worthington Belt-driven Class C Pump. Sizes 2-in. to 8-in.

size **impeller** only which can be adapted to different service conditions by changing the speed of the driver, and through it, the speed of the pump.

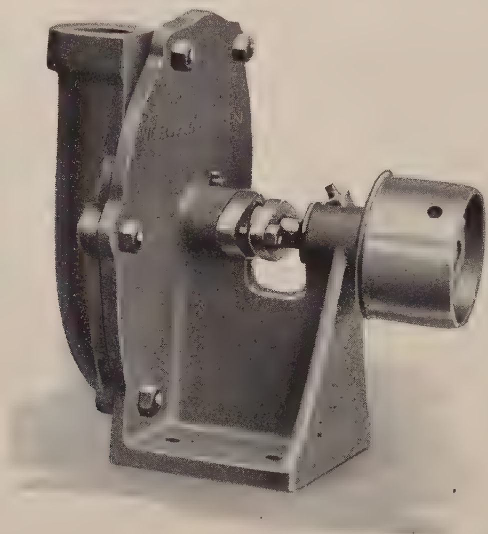


FIG. 17A. Worthington Belt-driven Class C Centrifugal Pump. Sizes 1-in. to 1½-in.

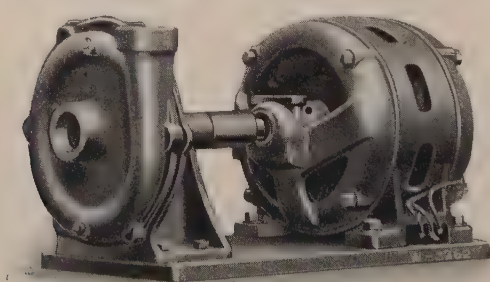


FIG. 17B. Worthington Class C Motor-driven Centrifugal Pump. Sizes 1-in. to 1½-in.

47. Unbalanced end thrust is taken care of by a ball-type thrust bearing. The **bearings** of all sizes are renewable. Both the inboard and outboard bearings of the 1 and 1½-in. and the inboard bearings of all sizes are grease lubricated. Sizes 2-in. and upward have outboard bearings of the pedestal type with automatic ring oiling.

48. TABLE OF CAPACITIES, SPEEDS AND HORSEPOWERS, VOLUTE PUMPS, CLASS C

Size of Pump	1 in.			1½ in.			2 in.		
Head in Feet	gal. per min.	hp.	r.p.m.	gal. per min.	hp.	r.p.m.	gal. per min.	hp.	r.p.m.
10	10	0.50	1100	18	0.50	725	53	1.00	625
	15	0.50	1200	30	0.50	735	67	1.00	660
	20	0.50	1300	50	0.60	785	78	1.00	695
	25	0.50	1390	70	0.70	880			
15	10	0.50	1310	20	0.70	870	65	1.00	770
	17	0.50	1390	40	0.75	920	82	1.00	810
	25	0.75	1550	60	0.90	980	96	1.25	850
	30	0.75	1670	80	1.10	1075			
20	10	0.75	1510	20	0.90	1000	75	1.50	885
	17	0.75	1570	45	1.00	1040	95	1.50	935
	25	0.75	1690	65	1.20	1110	111	1.75	980
	35	1.00	1900	85	1.50	1200			
25	10	0.75	1630	23	1.20	1115	84	1.75	990
	17	0.75	1690	50	1.30	1160	106	2.00	1050
	25	1.00	1840	75	1.60	1250	124	2.50	1100
	35	1.25	2080	95	1.85	1330			
30	10	0.75	1780	23	1.50	1220	91	2.00	1090
	17	1.00	1830	50	1.60	1250	116	2.75	1150
	25	1.25	1950	75	1.90	1340	135	3.00	1200
	35	1.30	2250	100	2.30	1430			
35	10	1.00	1880	25	1.80	1315	99	2.75	1170
	17	1.00	1940	50	1.90	1340	125	3.25	1240
	25	1.25	2060	80	2.30	1440	146	4.00	1290
	35	1.75	2370	110	2.90	1560			
40	10	1.25	2010	25	2.20	1410	105	3.25	1250
	17	1.25	2070	50	2.30	1440	134	4.00	1320
	25	1.50	2250	85	2.75	1540	156	4.50	1390
	35	2.00	2500	115	3.40	1650			
50	10	1.50	2260	25	3.00	1575	118	4.50	1400
	17	1.75	2320	50	3.20	1615	150	5.50	1480
	25	2.00	2460	85	3.60	1685	175	6.50	1550
	30	2.50	2600	120	4.40	1810			
60	10	2.00	2460	25	3.70	1720	129	5.75	1540
	15	2.25	2500	50	3.80	1735	164	7.00	1620
	20	2.50	2600	90	4.40	1810	191	8.50	1700
				130	5.50	1960			
70	......	....	......	30	4.50	1850	140	7.50	1660
	......	....	......	60	4.80	1860	177	9.00	1750
	......	....	......	100	5.70	1970	207	10.50	1830
	......	....	......	140	7.00	2120			
80							149	9.00	1780
							190	11.00	1870
							220	13.00	1960

48. TABLE OF CAPACITIES, SPEEDS AND HORSEPOWERS, VOLUTE PUMPS, CLASS C—Continued

Size of Pump	2½ in.			3 in.			4 in.		
Head in feet	gal. per min.	hp.	r.p.m.	gal. per min.	hp.	r.p.m.	gal. per min.	hp.	r.p.m
10	75	1.00	540	125	1.50	500	224	2.00	428
	98	1.25	560	165	1.75	535	304	2.25	460
	115	1.25	590	191	2.00	555	332	2.50	475
15	92	1.25	660	152	2.00	612	274	3.00	522
	120	1.50	685	200	2.25	655	373	3.50	560
	141	1.75	720	234	2.50	680	405	4.00	580
20	106	1.75	760	176	3.00	710	316	4.00	605
	140	2.00	785	232	3.25	760	430	5.25	650
	163	2.50	835	270	3.75	780	470	5.75	670
25	119	2.25	850	197	3.50	790	353	5.50	675
	156	2.50	880	260	4.25	845	482	6.50	725
	182	3.00	930	302	5.00	875	525	7.50	750
30	130	2.75	930	216	4.50	865	388	6.75	740
	170	3.25	965	285	5.25	930	528	9.00	795
	200	4.00	1020	330	6.25	960	575	9.50	820
35	140	3.25	1010	233	5.25	935	418	8.50	800
	184	4.25	1040	308	6.50	1000	570	10.75	860
	216	5.00	1100	360	7.50	1035	620	12.00	890
40	150	4.00	1075	250	6.50	1000	446	10.00	850
	197	5.00	1110	330	8.00	1070	610	13.50	920
	230	5.75	1180	380	9.25	1110	665	14.00	950
50	168	5.30	1200	280	8.50	1130	500	13.50	955
	220	6.75	1240	370	11.00	1200	680	17.50	1025
	258	8.00	1320	428	12.50	1240	745	19.00	1060
60	184	7.00	1320	305	11.00	1220	546	17.50	1045
	242	8.50	1360	402	13.50	1310	746	23.00	1120
	282	10.50	1440	468	16.00	1350	810	25.00	1160
70	198	8.50	1420	330	13.50	1330	590	21.50	1130
	262	11.00	1470	435	17.00	1420	805	28.50	1210
	305	13.00	1560	505	20.00	1460	880	32.00	1250
80	212	10.50	1520	352	16.00	1400	630	26.00	1210
	278	13.00	1570	465	21.00	1515	860	35.00	1300
	326	16.00	1660	540	25.00	1560	940	39.00	1340

48. TABLE OF CAPACITIES, SPEEDS AND HORSEPOWERS, VOLUTE PUMPS, CLASS C—Concluded

Size of Pump	5 in.			6 in.			8 in.		
Head in Feet	gal. per min.	hp.	r.p.m.	gal. per min.	hp.	r.p.m.	gal. per min.	hp.	r.p.m.
10	332	3.00	360	434	3.50	322	780	5.50	270
	450	3.25	390	572	4.00	335	1025	6.75	290
	550	3.75	420	750	4.50	357	1200	7.50	302
15	406	4.25	445	530	5.00	394	955	8.50	330
	554	5.25	476	700	6.00	410	1260	10.50	354
	675	6.00	515	920	7.50	436	1460	12.00	370
20	470	6.00	510	611	7.00	455	1100	12.00	380
	640	7.50	550	810	8.50	474	1450	14.50	410
	780	9.00	600	1065	10.50	505	1690	16.50	430
25	525	7.75	570	685	9.00	510	1230	15.50	427
	715	10.00	615	905	11.00	530	1620	19.00	455
	870	11.75	670	1180	13.00	565	1890	22.00	478
30	575	10.00	625	750	11.00	560	1350	19.50	466
	784	12.50	675	990	13.50	580	1775	24.00	500
	950	15.00	730	1300	18.00	620	2070	28.00	524
35	620	11.80	675	810	13.50	605	1460	23.50	505
	845	15.50	725	1070	17.00	626	1920	29.50	540
	1030	18.50	785	1400	22.00	670	2240	34.00	566
40	664	14.00	720	865	16.00	645	1560	28.50	540
	905	18.50	780	1142	18.00	670	2050	35.00	578
	1100	22.00	840	1500	26.00	715	2400	41.00	605
50	740	19.00	805	968	20.50	720	1750	38.00	604
	1010	25.50	870	1280	27.00	750	2300	47.50	645
	1230	31.00	940	1680	35.50	800	2670	55.00	676
60	812	24.00	785	1060	27.00	790	1910	48.50	660
	1110	32.00	950	1400	35.00	820	2500	61.50	706
	1350	40.00	1030	1840	46.00	875	2920	73.00	740
70	880	31.00	955	1145	34.00	850	2060	60.00	715
	1200	41.50	1030	1515	43.00	885	2710	76.00	763
	1455	51.00	1110	1980	57.50	945	3160	92.00	800
80	940	37.00	1020	1225	41.00	910	2210	73.00	763
	1280	50.00	1099	1620	53.00	950	2900	93.00	815
	1540	61.00	1190	2125	71.00	1020	3400	112.00	855

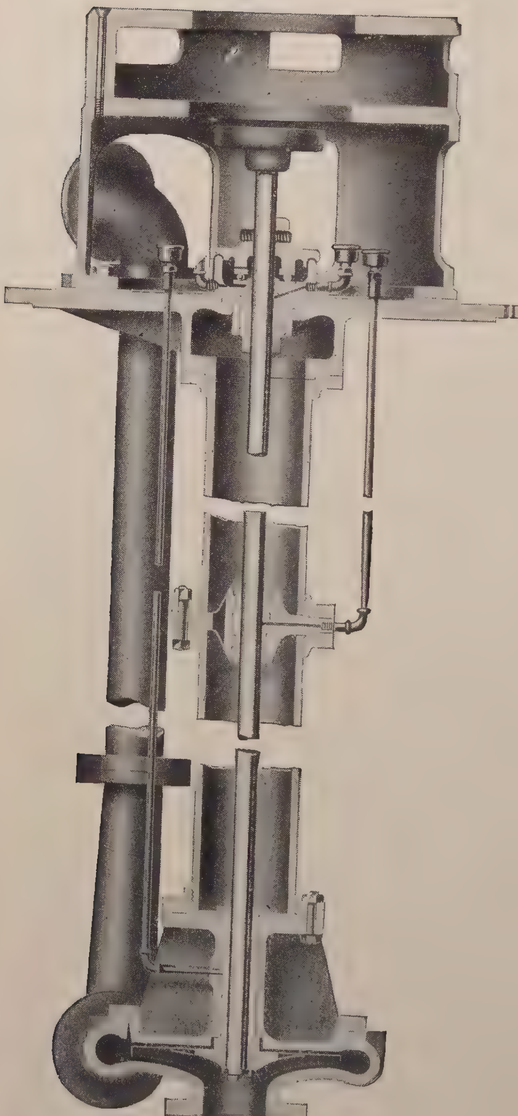


FIG. 18.
CROSS SECTION OF WORTHINGTON CLASS C OPEN-IMPELLER SUMP PUMP SIZES 1½-IN. TO 4-IN.

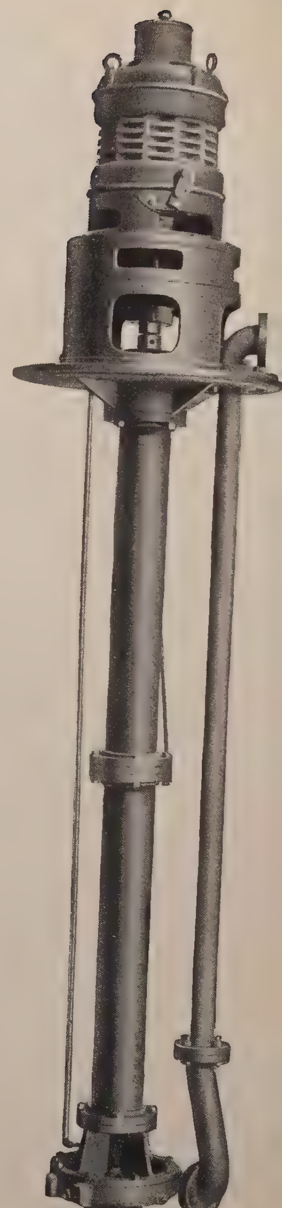


FIG. 18A.
GENERAL ELEVATION OF WORTHINGTON CLASS C OPEN-IMPELLER SUMP PUMP. DRIVEN BY VERTICAL MOTOR. SIZES 1½-IN. TO 4-IN.

49. STANDARD SUMP-PUMP RATINGS

Size of pump	1½ in.			2 in.			2½ in.			3 in.			4 in.		
Head Feet	gal. per min.	hp.	r.p.m.	gal. per min.	hp.	r.p.m.	gal. per min.	hp.	r.p.m.	gal. per min.	hp.	r.p.m.	gal. per min.	hp.	r.p.m.
10	60	1	830–1150	90	1	700–1150	140	2	660–1150	170	2	550– 900	350	3	500– 720
										250	3	650– 900			
15	75	1	1050–1250	80	1	810–1150	120	2	750–1150	230	3	700– 900	500	5	625–1150
				100	2	860–1150	185	3	815–1150						
20	60	1½	1100–1350	120	2	980–1150	140	2	780–1020	325	5	850–1150	450	5	620–1150
	100	2	1260–1550	150	3	1150–1500	165	3	900–1150						
25	45	1½	1150–1400	130	3	1100–1400	240	5	1050–1150	175	3	950–1150	500	7½	750– 900
	100	2	1340–1720							300	5	875–1150			
30	60	2	1300–1600	115	3	1150–1500	140	3	950–1150	250	5	900–1150	400	7½	750– 900
	100	3	1440–1730				225	5	1050–1150						
35	30	2	1330–1725	160	5	1300–1750	200	5	1100–1150	300	7½	1000–1150	500	10	850–1050
	100	3	1475–1750												
40	40	3	1420–1720	140	5	1350–1750	190	5	1100–1300	250	7½	1000–1150	400	10	850–1050
	120	5	1650–1750												
50	100	5	1750	125	5	1450–1750									

Capacities given in table are the maximum that can be obtained with size motor referred to.

50. Sump Pumps.—Fig. 18 illustrate the Worthington Class C Sump Pump for draining sump pits and work of a similar character, and for gritty, dirty water. The casing and impeller are the same as for corresponding size Class C volute pump. The column pipe encloses the shaft, supports the pump casing and intermediate bearings. The pump with its column and discharge pipe is securely bolted to and suspended from a heavily ribbed manhole cover, making the entire unit self-contained.

51. These units are arranged for submerged operation. All bearings are lubricated from compression grease cups located above the floor line. All pumps are provided with ball thrust bearings and are arranged for motor drive. Sump pumps can be arranged for automatic operation by installing the proper float switch and water-level control. For sizes and ratings see table, par. 49.

52. Standard Class OS Volute Pumps.—For general service conditions where the maximum head does not exceed 120 ft., the Worthington Class OS Volute with double-suction type impeller is highly recommended. This pump has been standardized in sizes from 2 to 12-in. inclusive. Sizes 2 to 6-in. inclusive are constructed with elbow-type suction nozzle and straight-vane impeller, as shown by Fig. 19. Sizes 8 to 12-in. inclusive are designed with volute-type suction nozzles.(Fig 20.) The impellers of the 8 and 10-in. sizes are of the straight-vane type except in special cases where the Francis mixed-flow type is employed. The 12-in. pump is always fitted with the Francis mixed-flow impeller.

53. The casing is of the open-throat type, split on the horizontal center line. The full diameter of the volute is employed, by limiting the diameter of the impeller so that it does not project into the casing. The whirling mass of liquid leaving the impeller acts with the whirling mass of liquid between the walls of the casing and of the impeller to form its own channel, resulting in uniform casing velocities without cross flows or eddies.

54. Casings of the open-throat type when used with full diameter impellers from heads of 130 to 150 ft. have an unbalanced and changing pressure in the space between the casing and impeller walls. This pressure not only varies in intensity but changes suddenly from one side of the impeller to the other giving a pound-

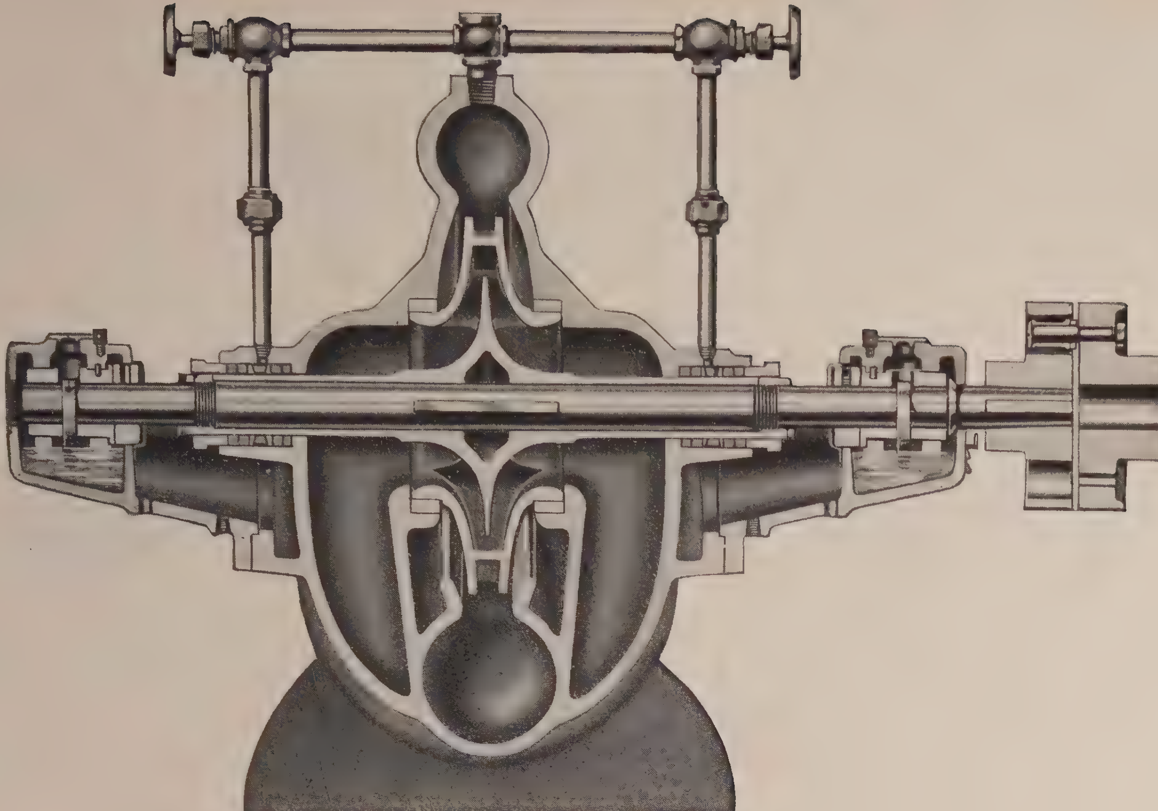


Fig. 19.
Cross Section of Worthington Double-Suction Class OS Volute Pump. Sizes 2-in. to 6-in.

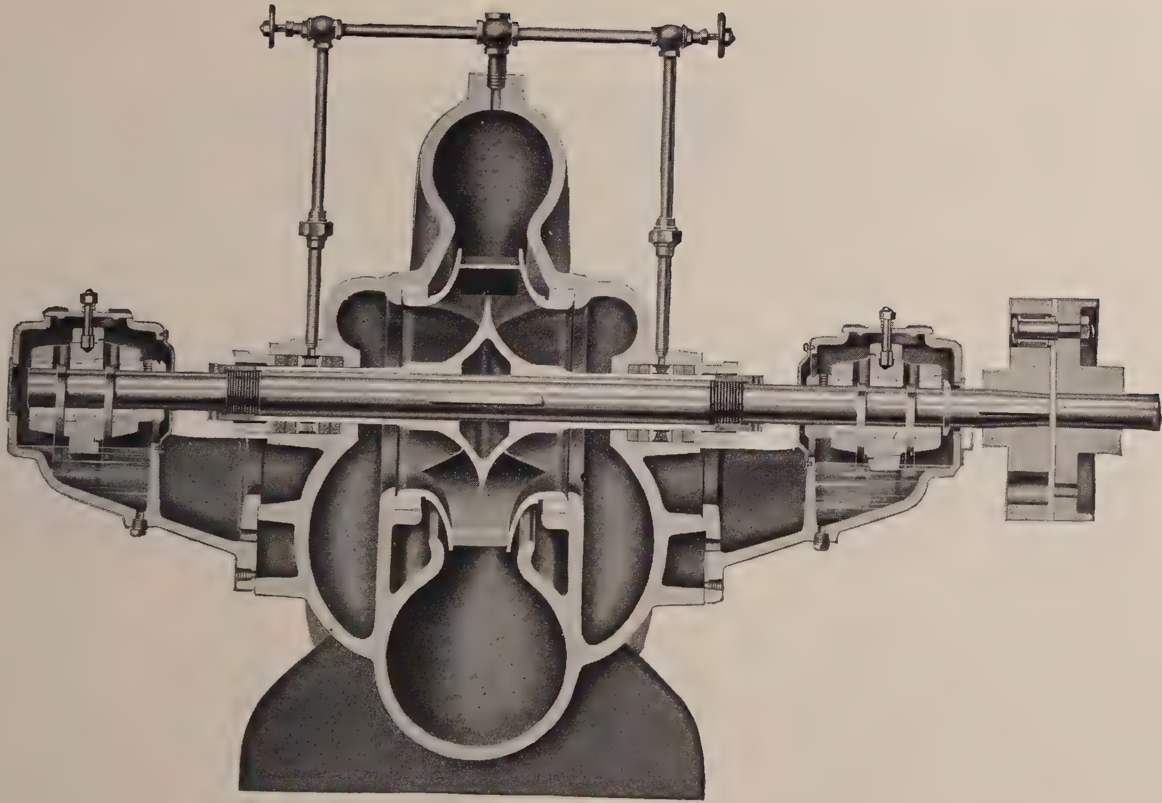


Fig. 20.
Cross Section of Worthington Double-suction Class OS Volute Pump. Sizes 8-in. to 12-in.

ing thrust that is harmful to thrust bearings of any type. This unbalanced condition does not exist at heads of 120 ft. and below and the pump is **hydraulically balanced.** Favorable conditions are sometimes encountered where small impeller diameters permit of heads up to 130 ft. without destroying the hydraulic balance. As the use of the Class OS pump is limited to such heads as permit of hydraulic balance, expensive thrust bearings of the marine or ball type are not needed.

55. The standard Class OS volute pump is exceptionally well designed and is built to the following specification:

Specifications—Worthington Class OS Volute Pumps

56. The **casing** is split on the horizontal center line with suction and discharge nozzles cast integral with the lower half. Access to the interior of the pump for the purpose of inspection or repair is obtained by removing the upper half of the casing. This may be done without disturbing the pipe connection or pump alignment. The joint between halves is sealed by an especially prepared oil gasket.

57. The **impeller** is of the double-suction enclosed type of high grade bronze with smooth finish. Two renewable bronze bushing rings with extra wide wearing surfaces maintain a close running fit with the impeller, thus preventing excessive leakage and maintaining the efficiency for long periods.

58. The **shaft** is of high quality steel of ample size, carefully machined and polished, and protected by removable bronze sleeves extending through the stuffing boxes. The sleeves are keyed to the shaft to prevent turning and are secured by an external lock nut and hold the impeller against lateral movement.

59. The **shaft bearings** are ring oiling, split horizontally, one located on each side of the pump. They consist of removable cast-iron babbitt-lined bushings carefully scraped to fit the shaft.

60. The **stuffing boxes** are of such size as to allow liberal packing. A lantern gland in each box is connected to the discharge water pressure, thus preventing suction air leaks.

61. The **bedplate** is of the box type built for stiffness and rigidity.

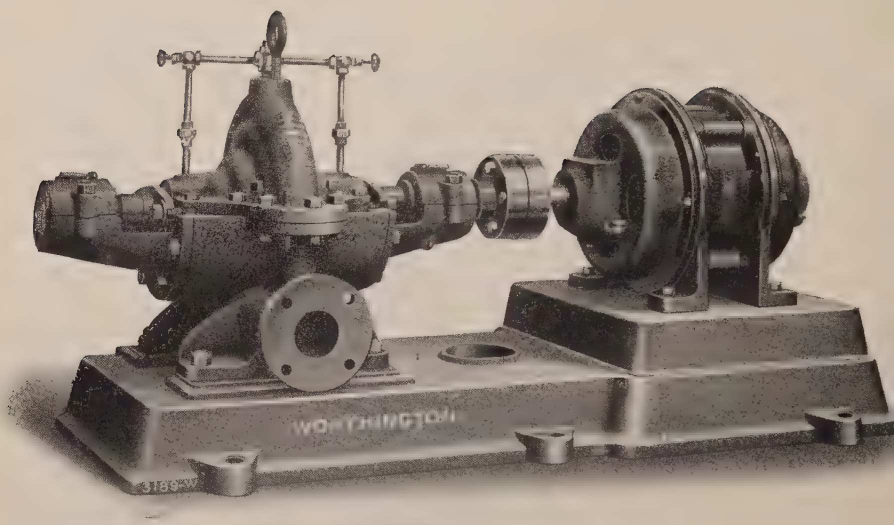


FIG. 21.

WORTHINGTON DOUBLE-SUCTION CLASS OS VOLUTE PUMP, MOTOR-DRIVEN

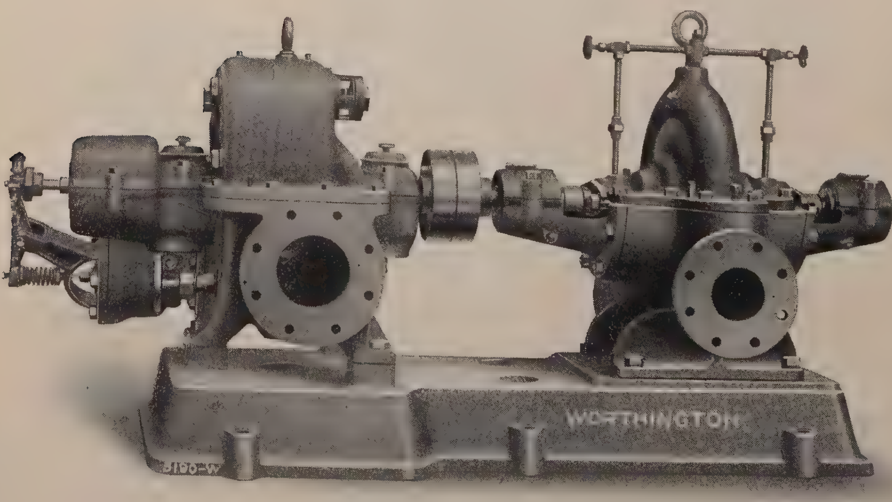


FIG. 22.

WORTHINGTON DOUBLE-SUCTION CLASS OS VOLUTE PUMP, STEAM-TURBINE DRIVEN

It has a flange at the bottom and is provided with suitable bosses for foundation bolts.

62. A flexible **coupling**, of the rubber-bushing and pin type, is supplied with each direct-connected unit.

63. The **equipment** of each pump includes the necessary special wrenches, water-seal piping, air and water cocks, glass oil gage, draw wire for water-seal cages, eye bolts and instruction bulletin, all contained in a tightly covered box.

64. Every part of the pump is of first class material and workmanship. All flanges are carefully faced and properly bolted. Both the exterior and interior of the pump are thoroughly painted before leaving the works.

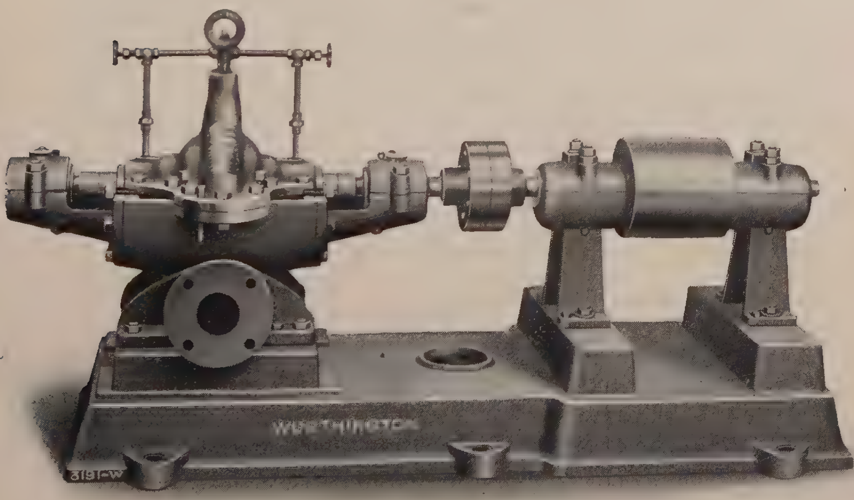


FIG. 23. Worthington Double-suction Class OS Volute Pump, belt drive.

65. RATINGS FOR OS PUMPS—60-CYCLE SPEEDS

Head in Feet	15 Feet		20 Feet		25 Feet		30 Feet		35 Feet	
Gal. per min.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.
50	1200	2	1200	2	1200	2	1200	2	1200	2
75	1200	2	1200	2	1200	2	1200	2	1200	2
100	1200	2	1200	2	1200	2	1200	2	1200	2
150	1200	3	1200	3	1200	3	1200	2	1200	2
200	1200	3	1200	3	1200	3	1200	3	1800	2
250	1200	4	1200	3	1200	3	1200	3	1200	3
300	1200	4	1200	4	1200	4	1200	4	1800	3
350	1200	4	1200	4	1200	4	1200	4	1200	4
400	1200	4	1200	4	1200	4	1200	4	1200	4
450	1200	5	1200	5	1200	5	1200	5	1800	4
500	900	6	1200	5	1200	5	1200	5	1200	5
600	900	6	1200	5	1200	5	1200	5	1200	5
700	900	6	1200	6	1200	6	1800	5	1800	5
800	1200	6	1200	6	1200	6	1200	6	1200	6
900	900	8	1200	6	1200	6	1200	6	1200	6
1000	900	8	900	8	1200	8	1200	6	1200	6
1100	900	8	900	8	1200	8	1200	8	1200	8
1250	1200	8	1200	8	1200	8	1200	8	1200	8
1500	900	10	900	10	1200	10	1200	8	1200	8
1750	900	10	1200	10	1200	10	1200	10	1200	10
2000	720	12	720	12	1200	10	1200	10	1200	10
2500	720	12	900	12	900	12	1200	12	1200	12
3000	720	12	900	12	900	12	1200	12	1200	12
3500	900	12	900	12	1200	12	1200	12	1200	12
4000	....	.....	......	.....	......	.....	1200	12	1200	12
4500	....	.....	......	.....	......	.....	1200	12	1200	12

NOTE: Speeds shown are synchronous; full-load speeds 4% less.

65. RATINGS FOR OS PUMPS—60-CYCLE SPEEDS—Continued

Head in Feet	40 Feet		45 Feet		50 Feet		60 Feet		70 Feet	
Gal. per min.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.
50	1200	2	1200	2	1200	2	1800	2	1800	2
75	1200	2	1200	2	1200	2	1800	2	1800	2
100	1200	2	1200	2	1200	2	1800	2	1800	2
150	1200	2	1200	2	1800	2	1800	2	1800	2
200	1800	2	1800	2	1800	2	1800	2	1800	2
250	1800	3	1800	3	1800	3	1800	3	1800	3
300	1800	3	1800	3	1800	3	1800	3	1800	3
350	1200	4	1200	4	1200	4	1800	3	1800	3
400	1200	4	1200	4	1200	4	1800	4	1800	4
450	1800	4	1800	4	1800	4	1800	4	1800	4
500	1200	5	1200	5	1200	5	1800	5	1800	5
600	1800	5	1800	5	1800	5	1800	5	1800	5
700	1800	5	1800	5	1800	5	1800	5	1800	5
800	1200	6	1200	6	1800	5	1800	5	1800	5
900	1200	6	1200	6	1800	5	1800	5	1800	5
1000	1200	6	1200	6	1800	6	1800	6	1800	6
1100	1200	8	1200	8	1200	8	1800	6	1800	6
1250	1200	8	1200	8	1200	8	1800	6	1800	6
1500	1200	8	1200	8	1800	8	1800	8	1800	8
1750	1200	10	1200	10	1800	8	1800	8	1800	8
2000	1200	10	1200	10	1200	10	1800	8	1800	8
2500	1200	10	1200	10	1200	10	1800	10	1800	10
3000	1200	12	1200	12	1200	12	1800	12	1800	10
3500	1200	12	1200	12	1200	12	1800	12	1800	10
4000	1200	12	1200	12	1200	12	1800	12	1800	12
4500	1200	12	1200	12	1800	12	1800	12	1800	12

NOTE: Speeds shown are synchronous; full-load speeds 4% less.

65. RATINGS FOR OS PUMPS—60-CYCLE SPEEDS—Concluded

Head in Feet	80 Feet		90 Feet		100 Feet		110 Feet	
Gal. per min.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.
50	1800	2	1800	2	1800	2	1800	2
75	1800	2	1800	2	1800	2	1800	2
100	1800	2	1800	2	1800	2	1800	2
150	1800	2	1800	2	1800	2	....	......
200	1800	2	1800	2	1800	2	....	......
250	1800	3	1800	3	1800	3	1800	3
300	1800	3	1800	3	1800	3	....	......
350	1800	3	1800	3	....	......	....	......
400	1800	4	1800	4	1800	4	1800	4
450	1800	4	1800	4	1800	4	1800	4
500	1800	5	1800	4	1800	4	1800	4
600	1800	5	1800	5	1800	5	1800	5
700	1800	5	1800	5	1800	5	1800	5
800	1800	5	1800	5	1800	5	1800	5
900	1800	5	1800	5	1800	5	....	......
1000	1800	6	1800	6	1800	6	1800	6
1100	1800	6	1800	6	1800	6	1800	6
1250	1800	6	1800	6	1800	6	1800	6
1500	1800	6	1800	6	1800	6	1800	6
1750	1800	8	1800	8	1800	8	1800	8
2000	1800	8	1800	8	1800	8	1800	8
2500	1800	10	1800	10	1800	10	1800	10
3000	1800	10	1800	10	1800	10	1800	10
3500	1800	10	1800	10	1800	10	1800	10
4000	1800	12	1800	10	1800	10	....	......
4500	1800	12	....	......	....	......	....	......

NOTE: Speeds shown are synchronous; full-load speeds 4% less.

WORTHINGTON MULTI-STAGE PUMPS

66. Standard Class SD Volute Two-Stage.—The standard pumps discussed in the preceding pages have all been of the single-stage double-suction type for large capacities at moderate heads. The class SD pump which will now be taken up for discussion is of the two-stage and four-stage single-suction type for relatively small capacities at from 175 to 600 ft. head. (See Fig. 28.)

67. The question naturally arises, Why are the single-stage pumps limited to moderate heads? In the paragraphs on design of impellers and their the effect of the higher specific speeds was discussed, and it was said that the measure of capacity of an impeller was the area through the passages. The higher-head impellers are naturally of the lower specific-speed type as the width of the passages necessary for the smaller capacities does not allow the use of the mixed-flow Francis vane necessary in the high specific speed impellers. The limiting factor in the use of low specific speed impellers is a function of the disk friction of the impeller walls. At constant speed, the disk friction will vary practically as the fifth power of the diameter, while the head produced varies only as the second power of the diameter, or peripheral, speed. Area through the impeller depends to some extent on the diameter so that a limit is reached where the narrow passages through the impeller, taken in conjunction with the larger disk friction, will cause a rapid decrease in the efficiency of the pump.

68. In order to obtain the best efficiencies for small capacities at heads of 175 to 350 ft. at speeds of 1800 r.p.m. and below, it is necessary to divide the head into a number of stages, preferably two; as a single-stage impeller cannot be designed with any degree of efficiency without resorting to the use of drivers of abnormally high speed.

69. Dividing the head into a number of stages improves the pump performance for high heads, and where only two stages are necessary the Worthington Class SD (Fig. 25,) with two single-suction impellers placed back to back will be found to be an unusually efficient compact pump.

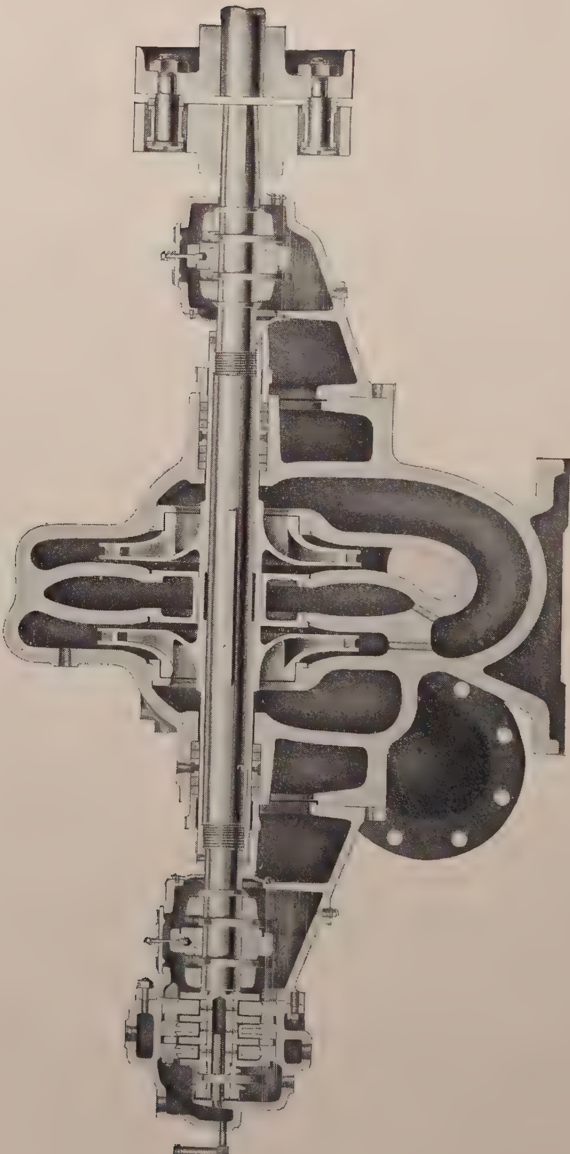


Fig. 24.

Cross Section of Worthington Standard Class SD Two-Stage Volute Pump.

70. The **casing** is of the open-throat volute form somewhat modified by being elongated. The volute of the first stage terminates in a cross-over port forming the suction nozzle of the second stage. Both the suction and the discharge nozzles are brought to one side of the casing.

71. The **impellers** are of the straight-vane single-suction type placed back to back with only one sealing ring for each impeller. A suitable **bushing** is inserted between the stages to prevent leakage from one to the other.

72. The **unbalanced pressure** created by the use of only one sealing ring is due to the stage pressure acting on the back wall of the impeller and includes the area between the outside diameter of the hub and the inside diameter of the sealing ring. The second stage impeller is subjected to first-stage pressure in the eye and second-stage pressure on the back, creating an unbalanced condition equal to that existing in the first-stage impeller but in the opposite direction, resulting in a hydraulically balanced pump.

73. In discussing the Class OS pump it was pointed out that the use of an open-throat casing for heads of 150 ft. and above required a high-grade thrust bearing. As the maximum stage pressures of the

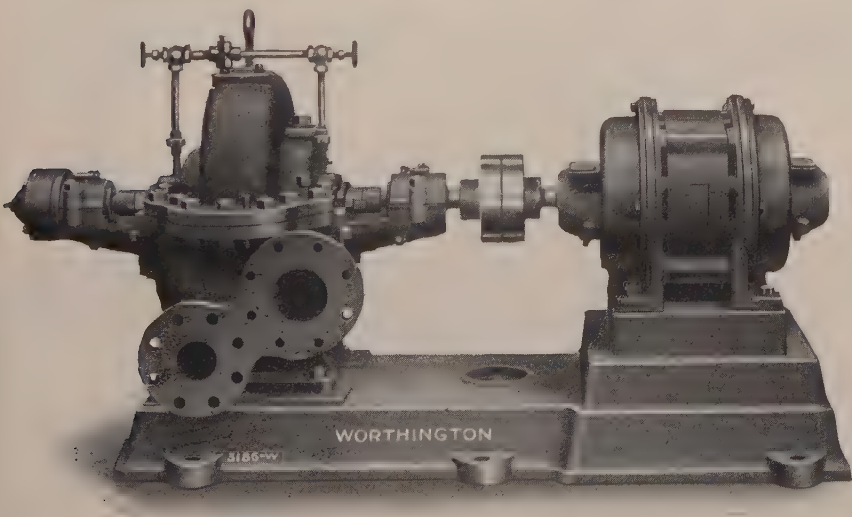


Fig. 25. Worthington Class SD Two-stage Volute Pump, motor-driven.

SD pumps are 175 ft. it is necessary to provide a suitable **thrust bearing** to take care of the unbalanced load. The SD volute pump is not subjected to a pounding thrust as the wall construction between stages is made so that the tendency to move toward one end will create a pressure on the impeller wall in the opposite direction.

74. The unbalanced thrust pressures in the 2 and 3-in. Class SD volutes are so small that a good ball-type thrust bearing is satisfactory for all requirements. The unbalanced pressures are somewhat greater in sizes 4 in. and upward, which are provided with water-cooled marine type thrust bearings with automatic lubrication.

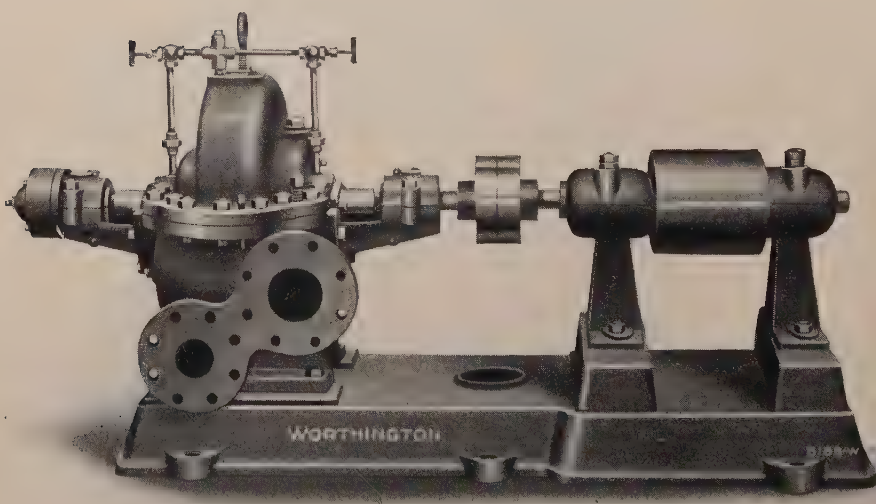


Fig. 26. Worthington Class SD Two-stage Volute Pump, belt drive.

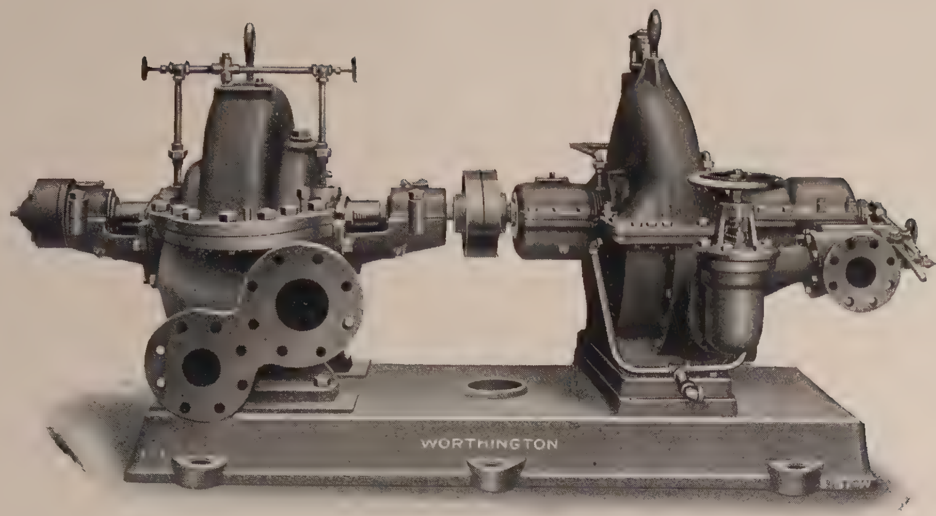


Fig. 27.

Worthington Class SD Two-stage Volute Pump, Steam Turbine-Driven

75. RATINGS FOR TWO-STAGE SD PUMPS—60-CYCLE SPEEDS

Head in Feet	190 Feet		200 Feet		210 Feet		220 Feet		230 Feet	
Gal. per min.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.
50	1800	2	1800	2	1800	2	1800	2	1800	2
75	1800	2	1800	2	1800	2	1800	2	1800	2
100	1800	2	1800	2	1800	2	1800	2	1800	2
125	1800	2	1800	2	1800	2	1800	2	1800	2
150	1800	2	1800	2	1800	2	1800	2	1800	2
175	1800	3	1800	3	1800	3	1800	3	1800	3
200	1800	3	1800	3	1800	3	1800	3	1800	3
225	1800	3	1800	3	1800	3	1800	3	1800	3
250	1800	3	1800	3	1800	3	1800	3	1800	3
275	1800	3	1800	3	1800	3	1800	3	1800	3
300	1800	3	1800	3	1800	3	1800	3	1800	3
350	1800	3	1800	3	1800	3	1800	3	1800	3
400	1800	3	1800	3	1800	3	1800	3	1800	3
450	1800	4	1800	4	1800	4	1800	4	1800	4
500	1800	4	1800	4	1800	4	1800	4	1800	4
550	1800	4	1800	4	1800	4	1800	4	1800	4
600	1800	4	1800	4	1800	4	1800	4	1800	4
700	1200	6	1200	6	1200	6	1200	6	1800	5
800	1200	6	1200	6	1200	6	1200	6	1200	6
900	1200	6	1200	6	1200	6	1200	6	1200	6
1000	1200	6	1200	6	1200	6	1200	6	1200	6
1100	1200	6	1200	6	1200	6	1200	6	1200	6
1250	1200	8	1200	6	1800	6	1800	6	1800	6
1500	1200	8	1200	8	1200	8	1200	8	1200	8
1750	1200	8	1200	8	1200	8	1200	8	1200	8
2000	1200	8	1200	8	1200	8	1200	8	1200	8
2500	....	....	....	....	....	....	....	....	....	....

NOTE: Speeds shown are synchronous; full-load speed 4% less.

75. RATINGS FOR TWO-STAGE SD PUMPS—60-CYCLE SPEEDS—Continued

Head in Feet	240 Feet		250 Feet		260 Feet		270 Feet		280 Feet	
Gal. per min.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.
50	1800	2	1800	2	1800	2	1800	2	1800	2
75	1800	2	1800	2	1800	2	1800	2	1800	2
100	1800	2	1800	2	1800	2	1800	2	1800	2
125	1800	2	1800	2	1800	2	1800	2	1800	2
150	1800	2	1800	2	1800	2	1800	2	....	....
175	1800	3	1800	3	1800	3	1800	3	1800	3
200	1800	3	1800	3	1800	3	1800	3	1800	3
225	1800	3	1800	3	1800	3	1800	3	1800	3
250	1800	3	1800	3	1800	3	1800	3	1800	3
275	1800	3	1800	3	1800	3	1800	3	1800	3
300	1800	3	1800	3	1800	3	1800	3	1800	3
350	1800	3	1800	3	1800	3	1800	3	1800	3
400	1800	3	1800	3	1800	3	1800	3	1800	3
450	1800	4	1800	4	1800	4	1800	4	1800	4
500	1800	4	1800	4	1800	4	1800	4	1800	4
550	1800	4	1800	4	1800	4	1800	4	1800	4
600	1800	4	1800	4	1800	4	1800	4	1800	4
700	1800	5	1800	5	1800	5	1800	5	1800	4
800	1200	6	1200	6	1800	6	1800	5	1800	5
900	1200	6	1800	6	1800	6	1800	6	1800	6
1000	1800	6	1800	6	1800	6	1800	6	1800	6
1100	1800	6	1800	6	1800	6	1800	6	1800	6
1250	1800	6	1800	6	1800	6	1800	6	1800	6
1500	1200	8	1200	8	1200	8	1200	8	1200	8
1750	1200	8	1800	8	1800	8	1800	8	1800	8
2000	1200	8	1800	8	1800	8	1800	8	1800	8
2500	1800	8	1800	8	1800	8	1800	8	1800	8

NOTE: Speeds shown are synchronous; full-load speed 4% less.

75. RATINGS FOR TWO STAGE SD PUMPS—60-CYCLE SPEEDS—Concluded

Head in Feet	290 Feet		300 Feet		325 Feet		350 Feet	
Gal. per min.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.	r.p.m.	Size in.
50	....	......	....	......	....	......	....	......
75	....	......	....	......	....	......	....	......
100	....	......	....	......	....	......	....	......
125	....	......	....	......	....	......	....	......
150	....	......	....	......	....	......	....	......
175	1800	3	1800	3	1800	3	1800	3
200	1800	3	1800	3	1800	3	1800	3
225	1800	3	1800	3	1800	3	1800	3
250	1800	3	1800	3	1800	3	1800	3
275	1800	3	1800	3	1800	3	1800	3
300	1800	3	1800	3	1800	3	1800	3
350	1800	3	1800	3	1800	3	1800	3
400	1800	3	1800	3	1800	3	1800	3
450	1800	4	1800	4	1800	4	1800	3
500	1800	4	1800	4	1800	4	1800	4
550	1800	4	1800	4	1800	4	1800	4
600	1800	4	1800	4	1800	4	....	......
700	1800	4	1800	4	....	......	....	......
800	1800	5	1800	5	1800	5	1800	5
900	1800	6	1800	6	1800	5	1800	5
1000	1800	6	1800	6	1800	6	1800	6
1100	1800	6	1800	6	1800	6	1800	6
1250	1800	6	1800	6	1800	6	1800	6
1500	1200	8	1200	8	1800	8	1800	8
1750	1800	8	1800	8	1800	8	1800	8
2000	1800	8	1800	8	1800	8	1800	8
2500	1800	8	1800	8	1800	8	1800	8

NOTE: Speeds shown are synchronous; full load speed 4% less.

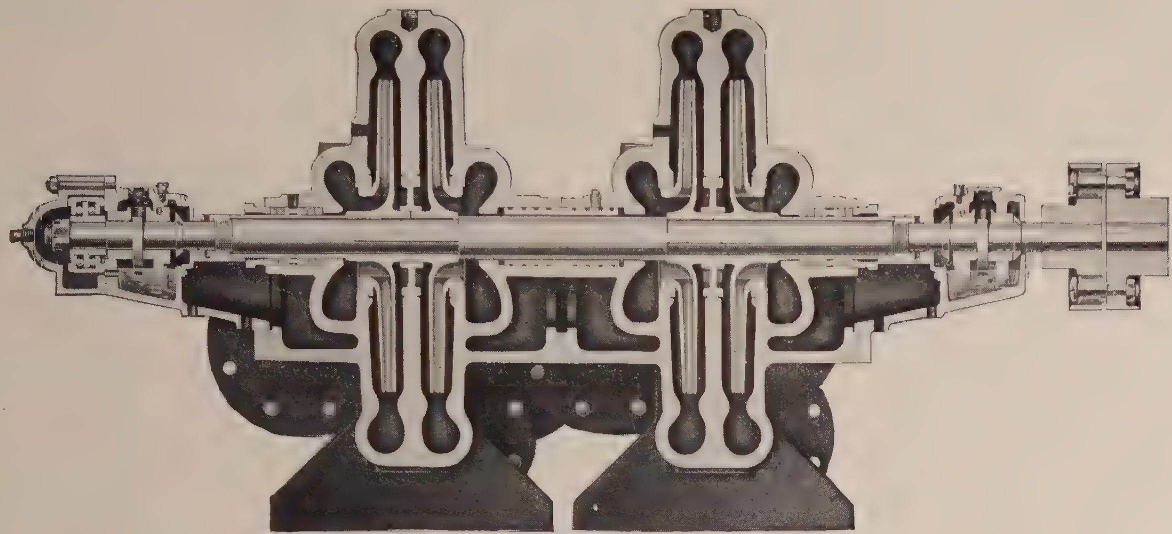


Fig. 28.

Cross section of Worthington Class SD Four-stage Volute Pump

76. Class SD Volute Four-Stage.—For conditions of service where small quantities of liquid are to be delivered against heads up to 600 ft. the Worthington Four-stage Class SD Volute can be used to advantage. Two standard two-stage casings are close coupled as shown by Fig. 28. The discharge nozzle of the first casing is connected to the suction nozzle of the second casing by means of a U-bend. The adjoining inside stuffing boxes provide the means of inserting a grooved stage bushing between the second and third stages. The grooves in this stage bushing are filled with fuse wire, forming a tight joint when the upper halves of the casing are bolted down.

77. A continuous shaft is furnished for mounting the impellers and other rotating parts. The bearings, shaft sleeves, etc. are interchangeable with corresponding parts of the two-stage pump. The four-stage construction is used on the small sizes and where it is impossible to obtain 175 ft. per stage. These four-stage pumps, either motor or steam-turbine driven, are ideal for **boiler-feed work** and are frequently used for this service.

78. Multi-Stage Pumps.—When service conditions require the use of a multi-stage pump in order to meet high heads, the pumping problem becomes complicated from both the hydraulic and the mechanical standpoint. To place three or more impellers in series operation means more than simply mounting them on a shaft and providing connecting passageways from one to the other.

79. In a multi-stage pump the more difficult problems are to transform efficiently the velocity head acquired by the liquid between stages into pressure, and to direct the flow of the liquid so as to avoid eddies and shocks. The hydraulic design of the multi-stage pump is otherwise practically the same as that of the single-stage pump.

80. Class JDS Pumps.—An efficient solution of these problems is found in the Worthington Class JDS Double-suction Multi-stage Pump shown by Fig. 29. The upper part of this cut illustrates a pump fitted with the small diameter impellers used with steam turbine drives with their high rotative speeds. The lower part shows larger diameter impellers for 1800-r.p.m. motor drive. The Class JDS pump is suitable for 100 lb. per sq. in. pressure per stage.

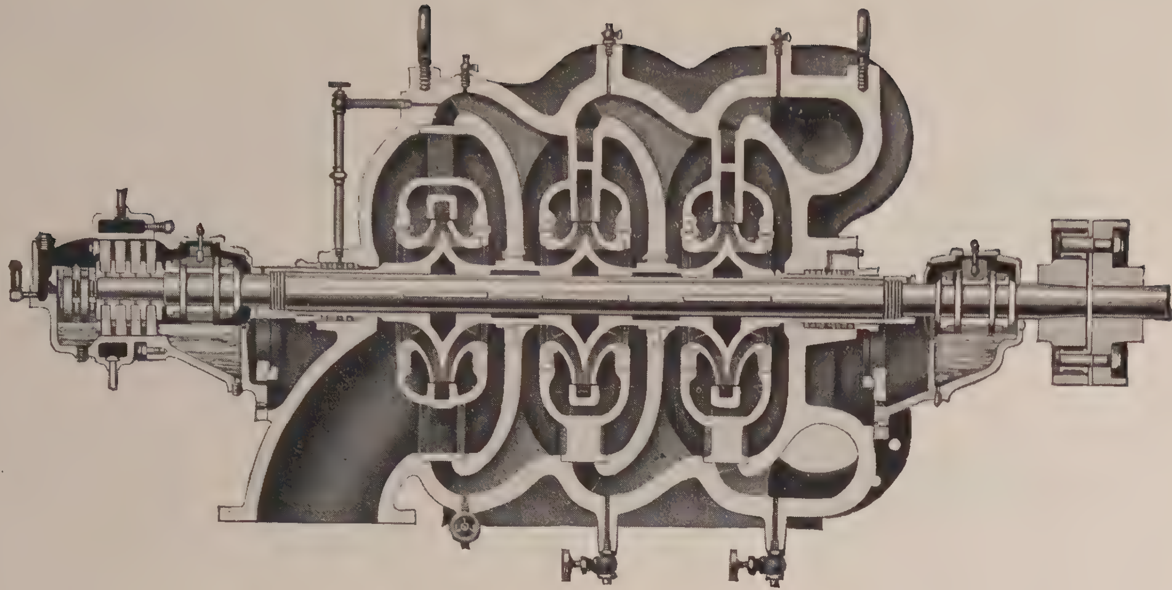


Fig. 29.

Cross Section of Worthington Class JDS Multi-stage Pump

Many five-stage units are now in operation at 450 lb. per sq. in pressure.

81. The **casing** is in halves with the suction and discharge nozzles cast integral with the lower half. The chambers for collecting the liquid as it is discharged from each diffusion ring are annular in form excepting that from the last stage, which is always of volute form. The passages or return channels leading from the discharge chamber of the next are cast into the casing. These return channels are accurately formed and with smooth surfaces to reduce skin friction to the minimum. The employment of diffusion rings permits of a simplified casing design of great rigidity and simplifies one of our mechanical problems.

82. In the Worthington Class JDS multi-stage pump the **diffusion rings** serve, first, as a guided passage for the liquid from the impeller to the casing and in which all head generation between stages is accomplished, and, second, as a chamber in which balanced pressures are obtained on the impeller walls.

83. Head generation depends to a certain extent on the proper velocity changes of the liquid. The passages in the diffusion ring through which the liquid flows to the casing are in effect a number of inverted venturi throats encircling the impeller. The area of these passages gradually increases toward the periphery of the diffusion ring and the velocity of the liquid flowing through them decreases as the area increases and builds up a pressure head.

84. To insure the **hydraulic balance** of the pump, the diffusion ring is designed so as to provide a balancing chamber on each side of the impeller. To equalize the pressures on the impeller walls, the balancing chambers are connected by means of a port in the diffusion ring. This construction is shown clearly by Fig. 29. Ports in the diffusion rings permit the passage of the liquid to the inside eye of the impeller. This is shown by the section of the first stage diffusion ring in Fig. 29. All liquid passages in the diffusion rings are smooth finished. The velocity pressure conversion throats are hand filed and scraped smooth to reduce loss by friction.

85. The double-suction type of **impeller** is used in the Class JDS pumps as it is more desirable than the single-suction both from the standpoint of low specific speeds and end thrust.

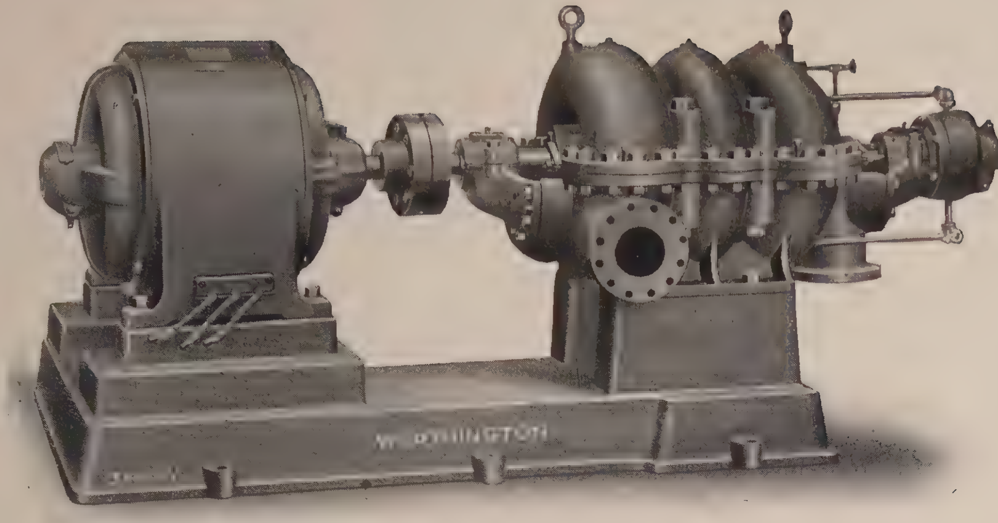


Fig. 30.

Worthington Class JDS Multi-stage Pump, Driven by Electric Motor.

86. The double-suction impeller takes half its capacity from each side; the single-suction takes its full capacity from one side, therefore for a given capacity the inlet diameter of the former type is considerably smaller than the inlet diameter of the single-suction type. For equivalent speed and head, the discharge diameter of both types would be the same.

87. The double-suction type, however, will have a greater ratio between the discharge diameter and the inlet diameter than the single-suction. This greater ratio of diameters gives the much desired increase in vane lengths, thus permitting the development of higher stage pressures with low specific speed impellers.

88. All **unbalanced thrust** resulting from high-stage pressures is taken care of by a well designed thrust bearing of the marine type with automatic lubrication.

SPECIFICATIONS—WORTHINGTON CLASS JDS MULTI-STAGE TURBINE PUMP

89. **Casing.**—The casing is split on the horizontal center line with suction and discharge nozzles cast integral with the lower half. Access to the interior of the pump for the purpose of inspection or repair is obtained by removing the upper half of the casing. This may be done without disturbing the pipe connection or pump alignment. The joint between halves is sealed by an especially prepared oil gasket.

90. **Impeller.**—The double-suction, enclosed-type impeller is used, which, located between balancing chambers, results in hydraulic balance in each stage. Two renewable bronze bushing rings maintain a close running fit with each impeller, thus preventing excessive leakage.

91. **Diffusion Ring.**—Each impeller discharges into a diffusion ring where the velocity head is converted into pressure head. These rings are split horizontally, each section being cast complete. Cored openings between the diffusion ports permit easy flow of water to both suction openings of the impeller. By removing one half of the ring, a free examination of the impeller can be made without disturbing the rotor setting.

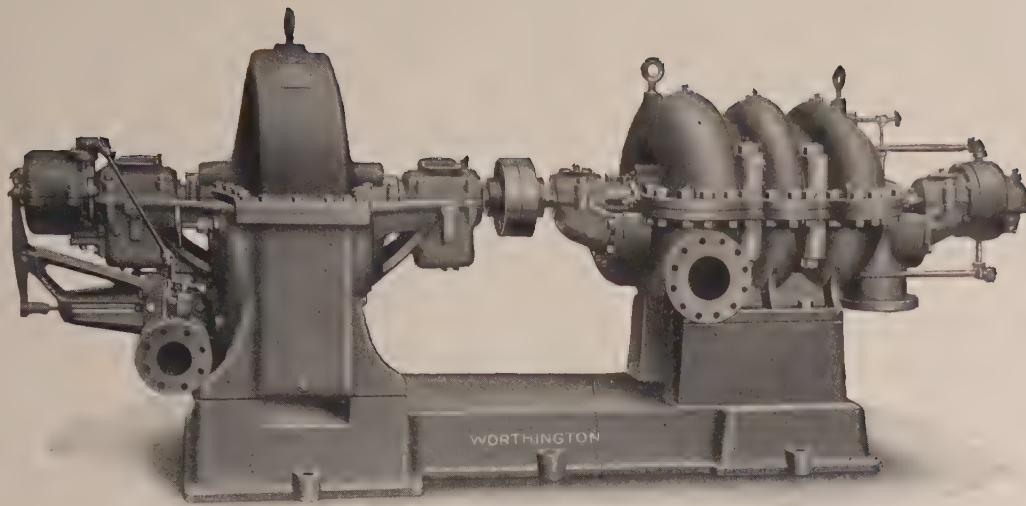


Fig. 31.

Worthington Class JDS Multi-stage Pump, Driven by Steam Turbine.

92. Shaft and Sleeves.—The shaft is of high quality steel of ample size, carefully machined and polished, and protected by removable bronze sleeves extending through the stuffing boxes. These sleeves are secured by an external lock nut and hold the impellers against lateral movement.

93. Bearings.—The shaft bearings are ring oiling, split horizontally, one located on each side of the pump. They consist of removable cast-iron babbitt-lined bushings scraped to fit the shaft.

94. A water-cooled, multi-stage, marine thrust bearing is provided to assure lateral alignments of impellers.

95. Stuffing Box.—The stuffing boxes are of such size as to allow liberal packing. A lantern gland in each box is connected to the discharge water pressure, thus preventing suction air leaks.

96. Bedplate.—The bedplate is of the box type properly ribbed for stiffness and rigidity. It has a flange at the bottom and is provided with bosses for foundation bolts.

97. Coupling.—A flexible coupling, of the rubber-bushing and pin type, is supplied with each direct-connected unit.

98. Equipment.—Each pump is equipped with the necessary special wrenches, water-seal piping, air and water cocks, glass oil gage, draw wire for water seal cages, eye bolts and instruction bulletin, all contained in a tightly covered box.

99. General Construction.—Every part of the pump is of first class material and workmanship. All flanges are carefully faced and properly bolted. Both the exterior and the interior of the pump are thoroughly painted before leaving the works.

WORTHINGTON BALL-BEARING CENTRIFUGAL PUMPS

100. To meet the demand for more efficient centrifugal pumps, Worthington has developed a line of ball-bearing pumps which are suitable for all kinds of general service.

101. The design of these pumps includes many novel features which

make for lower cost of operation and reduced maintainance.

102. Special care was taken to produce smooth flow through the water passages to minimize hydraulic losses.

103. The bearing construction assures permanent alignment of the rotor which not only improves the mechanical efficiency, but reduces wear.

104. For small capacities and high heads, the new pumps are designed to operate at a speed of 3600 r.p.m. This speed makes it possible to use impellers of smaller diameter, which can be cast in one piece.

105. The pump is, therefore, more rugged in construction and more efficient.

106. The Worthington Ball-bearing Pump is built in three types.

(1) Single-stage, single-suction	R and S
(2) Single-stage, double-suction	L, M and H
(3) Two-stage, single-suction	U

SPECIFICATIONS—WORTHINGTON R AND S VOLUTE PUMPS

107. **Casing.**—The casing is made of cast iron and is designed to produce smooth flow with gradual changes in velocity. It is split on the horizontal center line with the suction and discharge nozzles cast integral with the lower half. Access to the interior of the pump for the purpose of inspection or repair is obtained by removing the upper half of the casing. This may be done without disturbing the pipe connection or pump alignment. The joint between the halves is sealed by an oiled-paper gasket.

108. **Impeller.**—The impeller, which is of the single-suction enclosed type with the back acting as a hydraulic balance, is made of bronze. The hub of the impeller is of extra length and is firmly secured to the shaft with a feather key. Impeller bushing rings are made of special material for long life and are firmly secured in the casing by means of a tongue-and-groove fitting. The impeller is

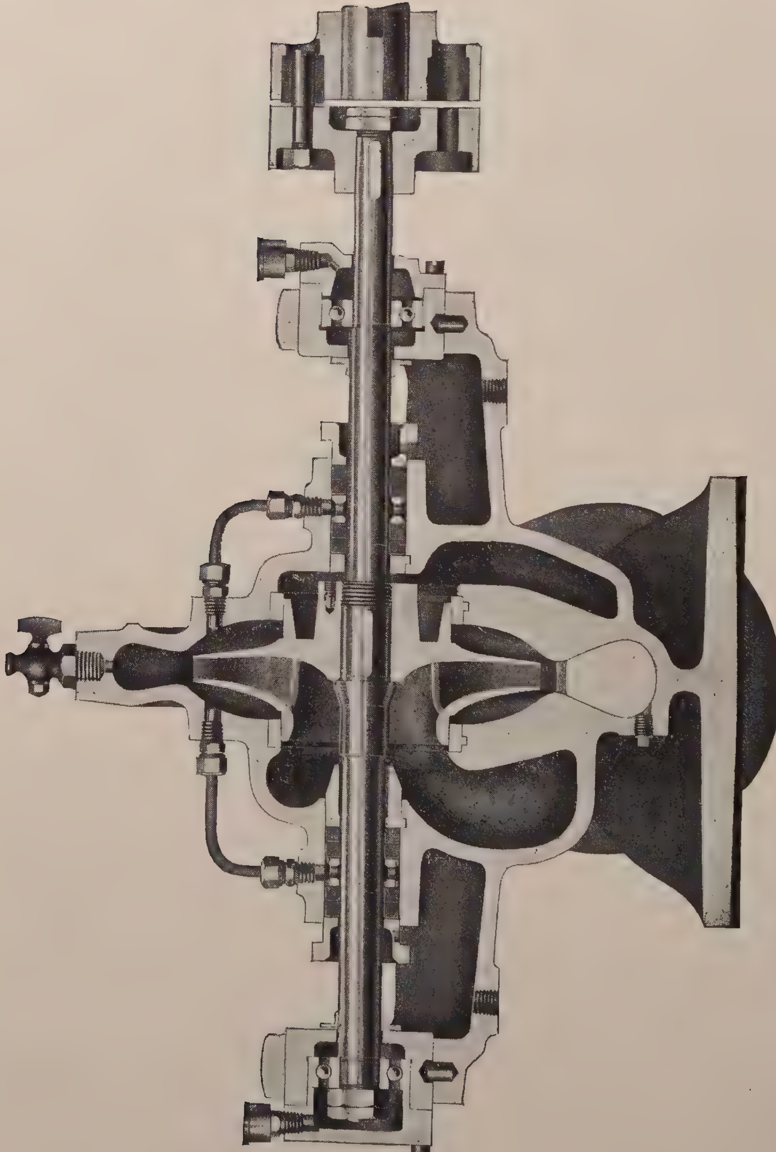


Fig. 32
Cross Section of Worthington R or S Volute Pump

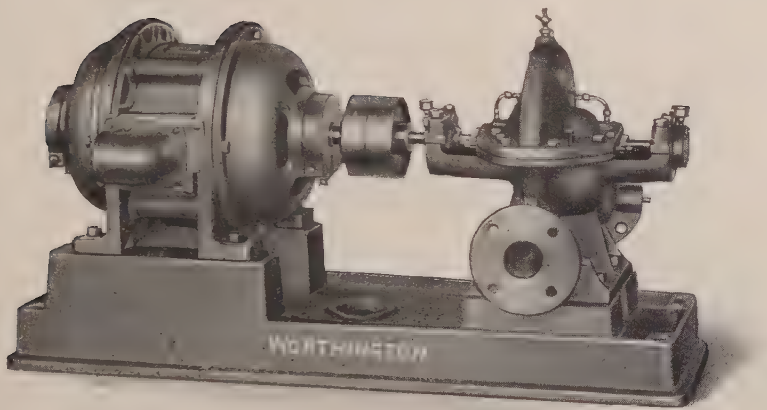


FIG. 33. Worthington R or S Volute Pump, Motor Drive

secured in position by means of a shaft nut holding it against a shoulder on the shaft.

109. Shaft.—The shaft is made of heat-treated steel. It is ground to accurate dimensions and polished to a smooth surface.

110. The shaft is supported on two Ball Bearings, one located on each side of the pump. The bearing housings are split at the center line, thus permitting easy removal of the bearings.

111. Stuffing Boxes.—The stuffing boxes are of extra depth to prevent air leakage. The glands are made of bronze and are secured with special swing bolts. A water seal gland is located in each stuffing box. A connection is made to the discharge of the pump to provide water under pressure for sealing purposes.

112. Bedplate.—The standard bedplate is of box-section and is provided with a raised lip. The top of the bedplate is sloped to drain off any leakage from the pump. The bedplate has a flange at the bottom and is provided with lugs for foundation bolts.

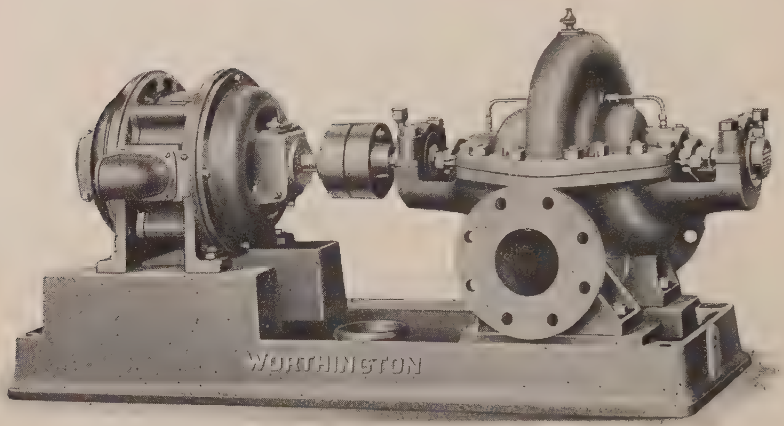


Fig. 34. Worthington H, L, M Volute Pump, Motor Drive

SPECIFICATIONS—WORTHINGTON H, L, AND VOLUTE PUMP

113. Casing.—The casing is made of cast iron and is designed to produce smooth flow with gradual changes in velocity. It is split on the horizontal center line with the suction and discharge nozzles cast integral with the lower half. Access to the interior of the pump for the purpose of inspection or repair is obtained by removing the upper half of the casing. This may be done without disturbing the pipe connection or pump alignment. The joint between the halves is sealed by an oiled-paper gasket.

114. Impeller.—The impeller, which is of the double-suction enclosed type, is made of bronze. The hub of the impeller is of extra length and is firmly secured to the shaft with a feather key. The impeller bushing rings are made of special material for long life and are firmly secured in the casing by means of a tongue-and-groove fitting.

115. Shaft.—The shaft is made of heat-treated steel. It is ground to accurate dimensions and polished to a smooth surface. The shaft sleeves of cast bronze protect the shaft at the stuffing boxes. The sleeves are secured in lateral position by external lock nuts and the impeller key is extended into the hub of the shaft sleeves.

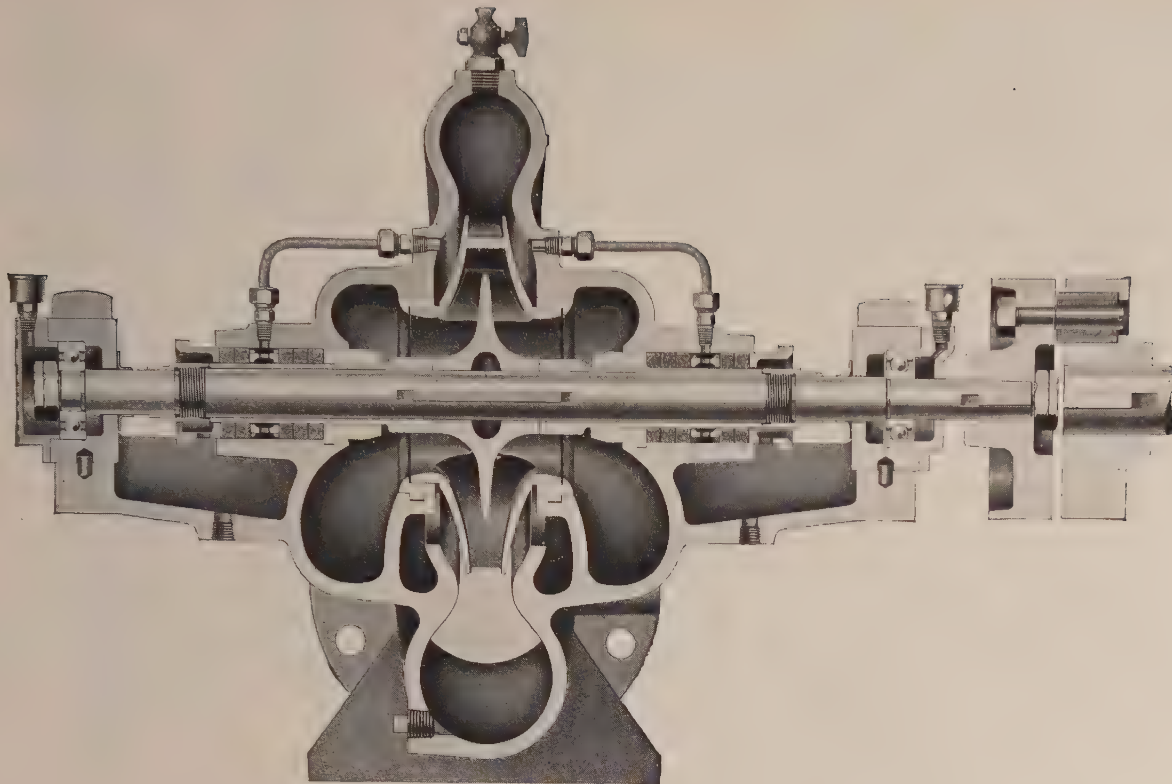


Fig. 34a
Cross Section of Worthington L, M or H Volute Pump

116. The shaft is supported on two Ball Bearings, one located on each side of the pump. The bearing housings are split at the center line, thus permitting easy removal of the bearings.

117. Stuffing Boxes.—The stuffing boxes are of extra depth to prevent air leakage. The glands are made of bronze and are secured with special swing bolts. A water-seal gland is located in each stuffing box. A connection is made to the discharge of the pump to provide water under pressure for sealing purposes.

118. Bedplate.—The standard bedplate is of box-section and is provided with a raised lip. The top of the bedplate is sloped to drain off any leakage from the pump. The bedplate has a flange at the bottom and is provided with lugs for foundation bolts.

119. Coupling.—A flexible coupling of the rubber-bushing-and-pin type is supplied with each direct-connected unit.

SPECIFICATIONS—WORTHINGTON U TWO-STAGE VOLUTE PUMPS

120. Casing.—The casing is made of cast iron and is designed to produce smooth flow with gradual changes in velocity. It is split on the horizontal center line with the suction and discharge nozzles cast integral with the lower half. Access to the interior of the pump for the purpose of inspection or repair is obtained by removing the upper half of the casing. This may be done without disturbing the

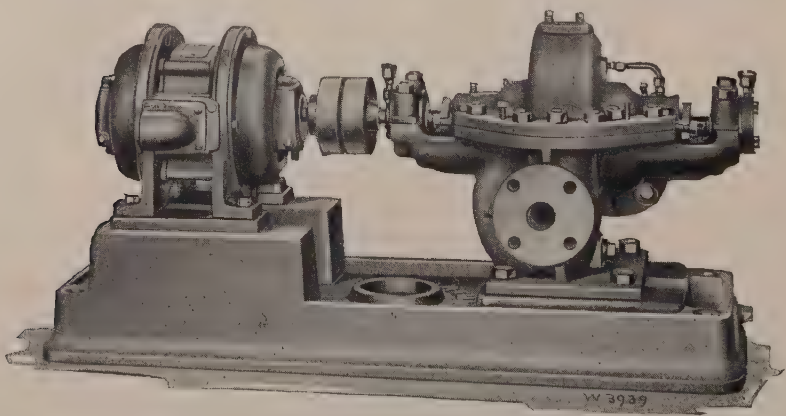


Fig. 35. Worthington U Two-stage Volute Pump

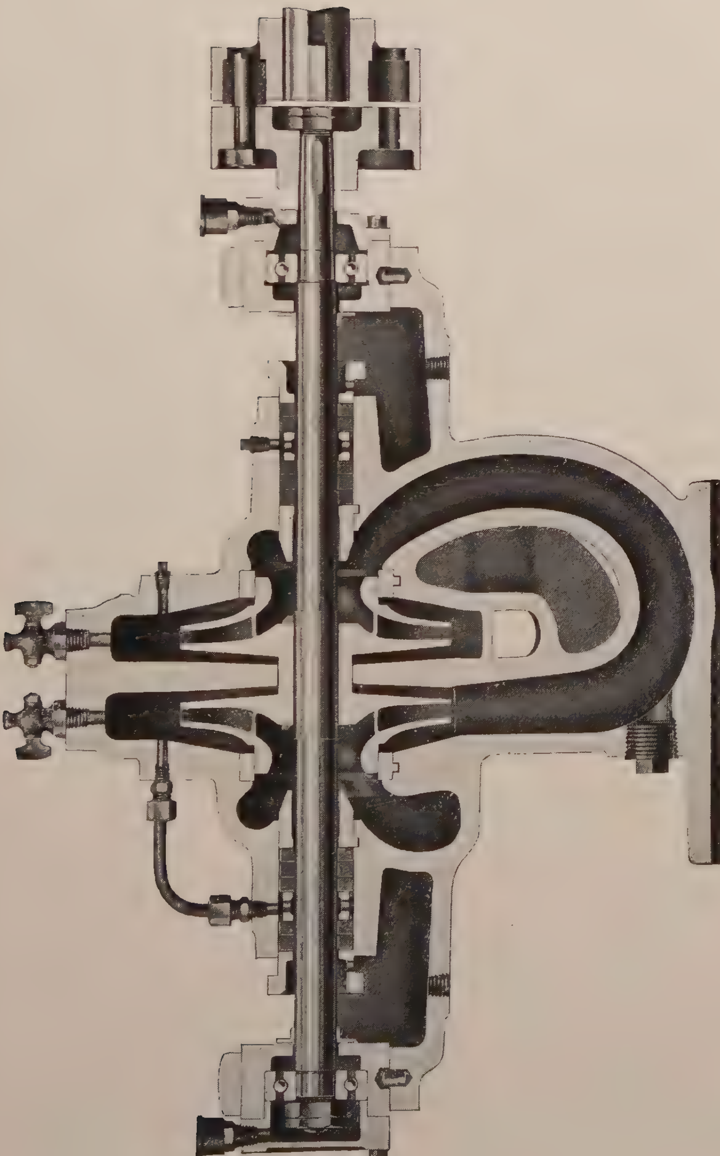


FIG. 35A.

CROSS SECTION OF WORTHINGTON U TWO-STAGE VOLUTE PUMP

pipe connection or pump alignment. The joint between the halves is sealed by an oiled-paper gasket.

121. Impellers.—The impellers are of the single-suction enclosed type and made of bronze. They will be opposed to one another, resulting in a hydraulically balanced pump. The hub of the impellers is of extra length and is firmly secured to the shaft with a feather key. Impeller bushing rings are made of special material for long life and are firmly secured in the casing by means of a tongue-and groove-fitting.

122. Shaft.—The shaft is made of heat-treated steel. It is ground to accurate dimensions and polished to a smooth surface.

123. The shaft is supported on two Ball Bearings, one located on each side of the pump. The bearing housings are split at the center line, thus permitting easy removal of the bearings.

124. Stuffing Boxes.—The stuffing boxes are of extra depth to prevent air leakage. The glands are made of bronze and are secured with special swing bolts. A water-seal gland is located in each stuffing box. A connection is made to the discharge of the pump to provide water under pressure for sealing purposes.

125. Bedplate.—The standard bedplate is of box section and is provided with a raised lip. The top of the bedplate is sloped to drain off any leakage from the pump. The bedplate has a flange at the bottom and is provided with lugs for foundation bolts.

126. Coupling.—A flexible coupling of the rubber-bushing-and-pin type is supplied with each direct-connected unit.

127. RATINGS FOR BALL-BEARING PUMPS, 60-CYCLE SPEEDS*

Head in Feet	20 Feet		30 Feet		40 Feet		50 Feet		60 Feet	
Gal. per min.	r.p.m.	size in.	r.p.m.	size in.	r.p.m.	size in.	r.p.m.	size in.	r.p.m.	size in.
50	1750	1½-R	1750	1½-R	1750	1½-R	1750	1½-R	1750	1½-U
100	1750	2½-S	1750	2½-S	1750	2½-S	1750	2½-R	1750	2½-R
150	1750	2½-S	1750	2½-S	1750	2½-S	1750	2½-R	1750	2½-R
200	.....	......	1150	3-L	1150	3-M	1750	2½-R	1750	2½-R
250	.....	......	1150	3-L	1150	3-M	1750	3-L	1750	3-L
300	.....	......	1150	3-L	1150	3-M	1750	3-L	1750	3-L
350	.....	......	1150	4-L	1150	3-M	1750	3-L	1750	3-L
400	1150	4-L	1150	4-L	1750	3-L	1750	3-L	1750	3-L
450	1150	4-L	1150	4-L	1750	3-L	1750	4-L	1750	3-L
500	1150	5-L	1150	4-L	1750	4-L	1750	5-L	1750	4-L
550	1150	5-L	1150	4-M	1750	4-L	1750	5-L	1750	4-L
600	1150	4-L	1750	4-L	1750	4-L	1750	4-L	1750	4-L
700	1150	4-L	1150	5-L	1750	4-L	1750	4-L	1750	5-L
800	1150	4-L	1150	4-L	1750	5-L	1750	4-L	1750	4-L
900	1150	6-L	1150	6-L	1750	4-L	1750	4-L	1750	4-L
1000	1150	6-L	1150	6-L	1750	5-L	1750	5-L	1750	4-L
1250	1150	6-L	1750	5-L	1750	5-L	1750	5-L	1750	5-L
1500	.....	......	1750	6-L	1750	6-L	1750	5-L	1750	5-L
1750	.....	......	1750	6-L	1750	6-L	1750	6-L	.....	......

Head in Feet	70 Feet		80 Feet		90 Feet		100 Feet		110 Feet	
50	1750	1½-U	1750	1½-U	.....	......	.....	......	.....	......
100	1750	2½-R	1750	2½-R	.....	......	3500	1½-R	3500	1½-R
150	1750	2½-R	1750	2½-R	1750	2½-R	1750	2½-R	3500	1½-R
200	1750	2½-R	1750	2½-R	1750	2½-R	1750	2½-R	1750	3-H
250	1750	3-L	1750	2½-R	1750	2½-R	3500	2½-S	1750	3-H
300	1750	3-L	1750	3-H	1750	3-H	3500	2½-S	1750	3-H
350	1750	3-L	1750	3-H	1750	3-M	1750	3-M	1750	3-H
400	1750	3-L	1750	3-M	1750	3-M	1750	3-M	1750	3-M

NOTE: Speeds shown are approximate full-load speeds.

*Concluded on following page.

127. RATINGS FOR BALL-BEARING PUMPS, 60-CYCLE SPEEDS

Head in Feet	70 Feet		80 Feet		90 Feet		100 Feet		110 Feet	
Gal. per min.	r.p.m.	size in.	r.p.m.	size in.	r.p.m.	size in.	r.p.m.	size in.	r.p.m.	size in.
450	1750	3-M	1750	3-M	1750	3-M	1750	3-M	1750	3-M
500	1750	3-M	1750	4-M	1750	3-M	1750	3-M	1750	3-M
550	1750	4-L	1750	4-M	1750	4-M	1750	3-M	1750	3-M
600	1750	4-L	1750	4-M	1750	4-M	1750	4-M	1750	4-M
700	1750	4-L	1750	4-M	1750	4-M	1750	4-M	1750	3-M
800	1750	4-L	1750	4-M	1750	4-M	1750	4-M	1750	4-M
900	1750	4-L	1750	4-M	1750	4-M	1750	4-M	1750	4-M
1000	1750	5-L	1750	5-L	1750	4-M	1750	4-M	1750	4-M
1250	1750	6-L	.....	......	.....	......	.....	......	.....	......
Head in Feet	120 Feet		140 Feet		160 Feet		180 Feet		200 Feet	
50	.....	......	3500	1½-U	3500	1½-U	3500	1½-U	3500	1½-U
100	3500	1½-R	3500	1½-R	3500	1½-U	3500	1½-R	3500	1½-R
150	3500	1½-R	3500	2½-S	3500	1½-R	3500	2½-S	3500	2½-R
200	1750	3-H	3500	2½-S	3500	2½-S	3500	2½-S	3500	2½-R
250	3510	2½-S	3500	2½-S	3500	2½-S	3500	2½-R	3500	2½-R
300	1750	3-H	3500	2½-S	3500	3-L	3500	2½-R	3500	2½-R
350	1750	3-H	1750	3-H	3500	3-L	3500	2½-R	3500	2½-R
400	1750	3-H	1750	3-M	3500	3-L	3500	3-L	3500	2½-R
450	1750	3-M	1750	3-M	3500	3-L	3500	3-L	3500	3-L
500	1750	3-M	1750	3-M	3500	3-L	3500	3-L	3500	3-L
550	1750	3-M	3500	3-L	3500	3-L	3500	3-L	3500	3-L
600	1750	3-M	3500	3-L	.....	......	3500	3-L	.....	......
700	1750	4-M	3500	4-L	.....	......	3500	3-L	.....	......
800	1750	4-M	3500	4-L	.....	......	.....	......	.....	......
900	1750	4-M	.....	......	.....	......	.....	......	.....	......
1000	3500	4-L	.....	......	.....	......	.....	......	.....	......

NOTE: Speeds shown are approximate full-load speeds.

SPECIAL-SERVICE WORTHINGTON PUMPS

154. Drainage Pumps.—The reclamation of marsh and swamp land for agricultural or other purposes led to the demand for a pump capable of delivering large quantities of water at extremely

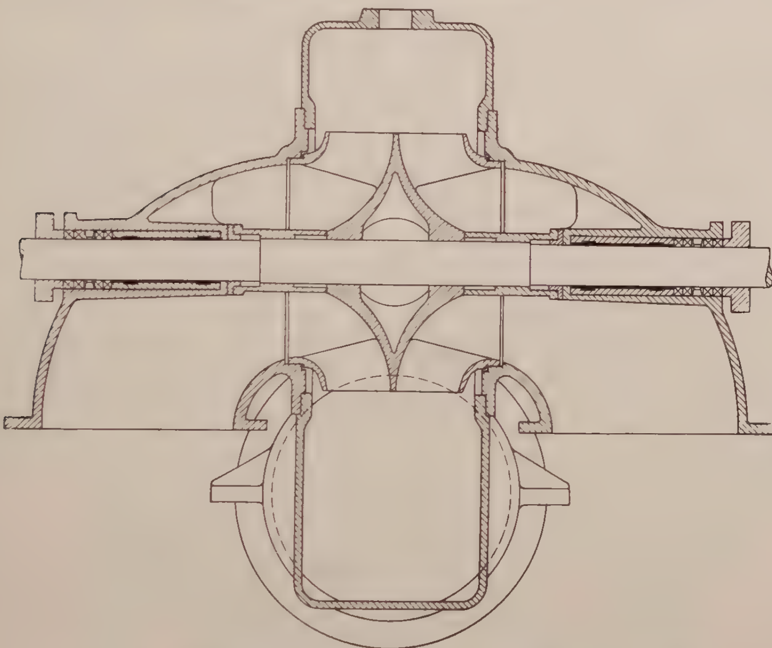


Fig. 36. Cross section showing general construction of Worthington Drainage Pump, with inside bearings, double suction.

low heads when driven by medium speed steam or oil engines. After a careful study of the requirements of this work, Worthington developed the medium-speed double-suction drainage pump in sizes from 16 to 48-in. inclusive. The amount of energy to be regained by velocity changes in a drainage pump is naturally so small on account of the low-head conditions that extra wide full area casing and impeller are required. Separate suction elbows are used to reduce entrance losses and to enable greater capacities to be obtained. The shaft bearings are built into the suction elbows of the smaller sizes. Independent shaft bearings of the pedestal type are used on the larger sizes. The general construction of the drainage pump is shown by Fig. 36. Both the casing and

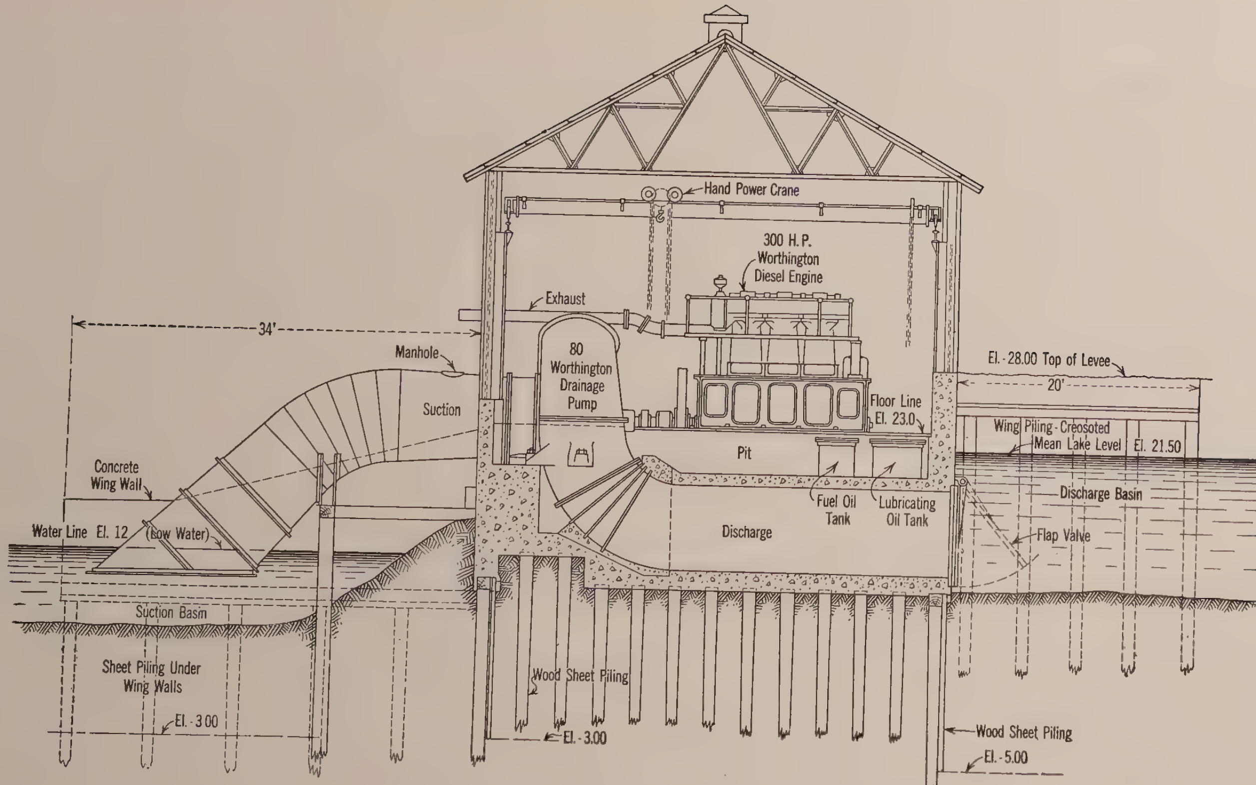


Fig. 37.

Typical Installation of Worthington Screw-impeller Tyye Drainage Pump, connected to Worthington Diesel Engine

suction elbows are split on the horizontal center line to facilitate examination and cleaning of the pump.

155. Fig. 37 illustrates an installation of drainage pumps. These pump houses are generally located on the top of a dike or dam with the suction basin on one side of the dike and the delivery basin on the other. The elevation of the water in both basins is below the floor line of the pump. Both the suction and discharge are under pressures less than atmospheric so that in effect a **siphon setting** is obtained.

156. The siphon setting eliminates valves in both suction and discharge and reduces entrance and exit losses to a minimum. The ends of the suction and discharge pipes are submerged. Both can be and should be properly expanded or bell mouthed for low entrance and velocity losses.

157. The pump being under a vacuum, no dirty water forces its way into the bearings when they are located in the suction elbows.

158. Screw Pump.—The **crude oil engine** with its advantage of low operating costs and low stand-by charges is an ideal prime

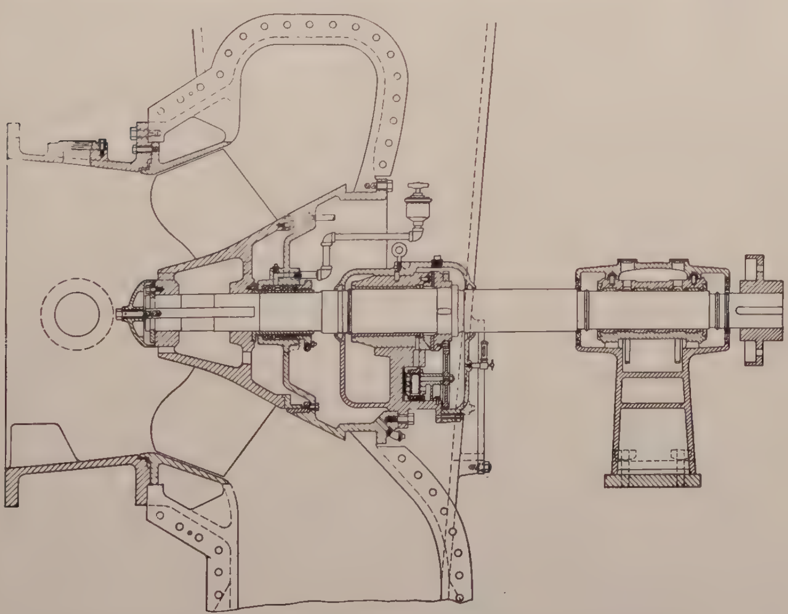


FIG. 38. Cross section showing general construction of Worthington Screw Pump for drainage and irrigation work.

mover for drainage and irrigation work. In the modern oil engine as developed by Worthington, the speed has been increased above that of the steam engine of corresponding horsepower as well as above the speeds of the older types of oil engine. This increase of speed in this type of prime mover called for the development of the Worthington Screw Pump (Fig. 38) for low-head (5-10 ft.)

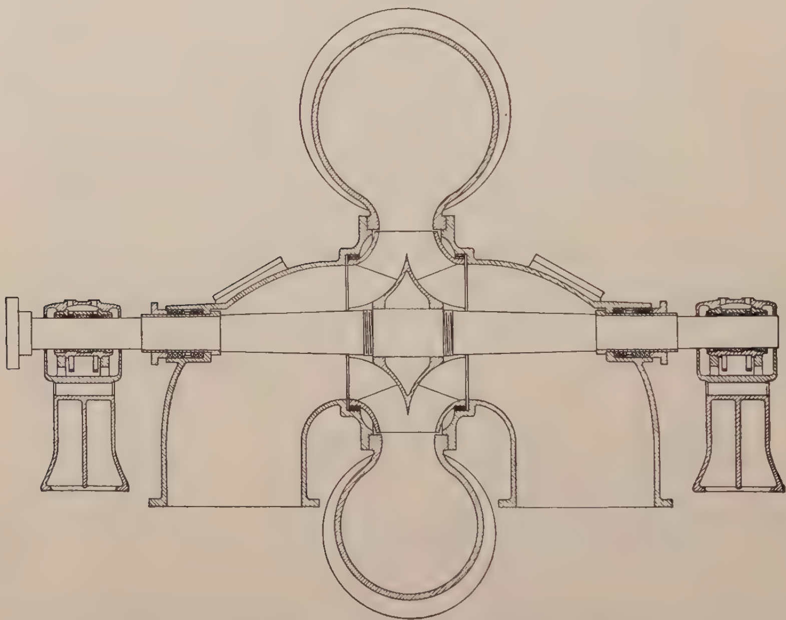


Fig. 39. Cross section showing general construction of Worthington Double-Suction, Low-head Pump with outside Bearings. For irrigation work.

drainage work. This, as previously mentioned, has the highest specific speed impeller as yet produced. Double-suction impellers have an equivalent speed 40 per cent higher than the single suction; the single-suction screw impeller exceeds the best that the double-suction can give at equivalent rotative speeds.

159. For drainage or irrigation work where the heads do not exceed 10 ft., the single-suction screw pump in sizes up to 80 in. with modern oil-engine drive, or with other prime movers at corresponding speeds, will give results that cannot be attained by any other type of pump. The double-suction screw pump, size

60 in. and upward, enables still larger capacities to be obtained with maintained oil-engine speeds.

160. The Worthington Single-suction Screw Pump is built in sizes from 20 to 80 in. inclusive. The design is similar to Fig. 38, which shows in profile the diagonal flow feature of the impeller and the apparent right angle turn of the water. The end suction is of special design for high specific speed impellers. The impeller or screw is located between this entrance and the volute spiral casing. The bad effects of the right-angle turn are minimized by designing the impeller so that the absolute path of the water spreads over a long arc as it advances, giving in effect a turn where the radius is extremely long. The distribution across the section is uniformly at a long radius, enabling the use of an impeller with almost axial flow, and which turns the water through 90 deg. without loss. (The flow through the impeller is slightly diagonal.) This construction maintains the full effect of centrifugal force resulting in full capacities and high efficiencies; 80 per cent on 5 ft. head having been obtained.

161. The early designers of axial-flow impellers which were inserted in a pipe did not take proper care of the elements of centrifugal force. This neglect of the fundamental principle of centrifugal pump design resulted in a considerable loss of both capacity and efficiency in these early pumps. These same fundamentals are overlooked by some designers today.

162. Pumps for Irrigation.—Closely allied with the problems of reclaiming waste marsh lands by drainage are the problems of reclaiming arid lands by irrigation. The pumping requirements are practically the same excepting that in irrigation work the heads range from 20 to 30 ft. as compared with the 5 to 10 foot range in drainage.

163. For driving the larger pumps for irrigation projects, steam engines or oil engines are usually employed. The higher heads require greater horsepower, resulting in lower speed engines and larger diameter pump impellers than are used in drainage work.

164. Pumps for irrigation work range in size from 30 to 60 in. The general design is shown by Fig. 39. This design is similar to the medium-speed drainage pump except that the bearings are located outside the suction elbows.

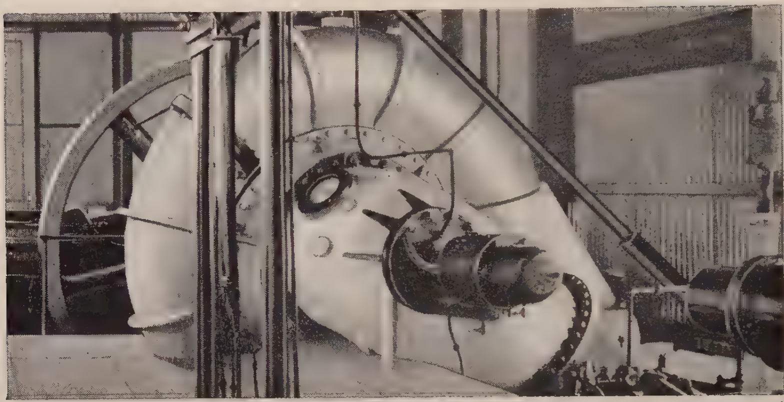


FIG. 40. Worthington 45-in. Double-Suction Irrigation Pump, direct-connected to tandem compound steam engine.

165. Fig. 40 shows a 45-in. irrigation pump direct-connected to a tandem-compound steam engine. The pump is located between the high and low-pressure engines. The impeller is mounted directly on the engine shaft, doing away with all couplings.

In some cases motor-driven pumps may be used, in others steam or oil-engine drives are advisable and others may require the use of a multi-stage high-head turbine pump. Worthington engineers will gladly aid customers in the solution of their problems.

166. Underwriter Fire Pumps.—The Worthington Centrifugal Underwriter Fire Pump is designed for 100 lb. per sq. in. pressure and is constructed to meet the exacting requirements of the Associated Factory Mutual Fire Insurance Companies., and the National Board of Fire Underwriters.

167. For 500 to 1500 gal. per min. the two-stage pump similar to the Class SD volute pump as illustrated and described in Par. 66, is used. For 1000 to 1500 gal. only, a single-stage volute pump is used.

168. The Worthington Underwriter Fire Pump embodies all the special features required by the Underwriters' specifications, including the use of non-corrosive metals for all working parts, accessibility of all running parts, hose manifolds, hose gates, water relief valve, waste cone, starting valve connections, vacuum gage and discharge pressure gage.

169. Worthington Underwriter Fire Pumps may be direct-connected through a flexible coupling to an electric motor, gasoline

engine or steam turbine. The pump and driver will always be mounted on a heavy, rigid cast-iron base plate. Fig. 41 shows a 1000-gal. Underwriter fire pump of the two-stage type direct-connected to an electric motor with a shield between the pump and motor to prevent any water from splashing on the motor.

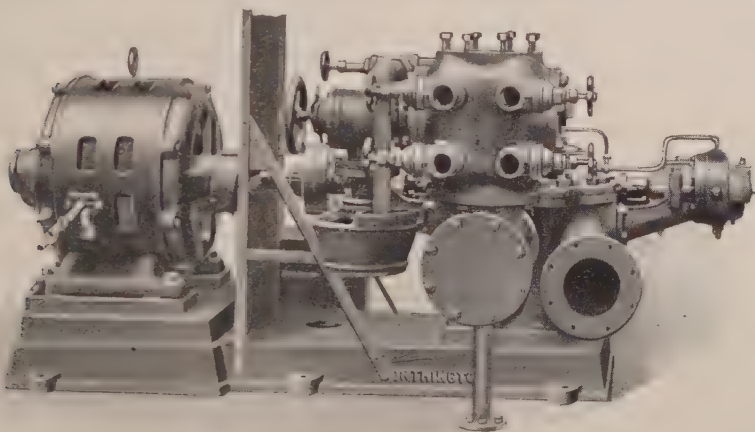


FIG. 41. Worthington Underwriter Fire Pumps, direct-connected to electric motor.

170. RATINGS FOR WORTHINGTON CENTRIFUGAL UNDERWRITER FIRE PUMPS

No. Stages	Capacities		Dia. Pipe Openings		40° Motor		Gasoline Engine	
	Gal. per min.	No. nozzle streams	Suc.	Disch.	hp.	Speed	hp.	Speed
2	500	2–1⅛″	6	6	60	1200–1750	75	1200
2	750	3–1⅛″	8	8	75	1150–1750	100	1200
2	1000	4–1⅛″	8	8	100	1150–1750	125	1200
2	1500	6–1⅛″	10	10	150	1150–1750	168	1200
1	1000	4–1⅛″	8	8	100	1750	Cannot be operated at gasoline engine speeds.	
1	1500	6–1⅛″	10	10	150	1750		

171. Special Pumps.—Both manufacturers and purchasers of pumping machinery realize that there are service conditions where pumps which have been developed as standard cannot be applied. Service conditions may fall just outside the limits of the standard, the nature of the liquid to be pumped may require special mechanical construction, or some other factor may necessitate the alteration of existing patterns or the development of a special pump for the service.

172. In their eighty-seven years, experience as manufacturers of pumping machinery, innumerable special problems have been analyzed and the proper pump designed, and placed in successful operation by Worthington. This field is entirely too broad to be covered fully, but a few cases will be cited where existing patterns have been altered to meet special conditions.

173. Open Impeller Double-Suction Pumps Class—The open-impeller volute pump, strictly speaking, may not be construed as special pumps and on the other hand they are not strictly a standard, as their field of application is not general but limited to the pumping of sewage, paper pulp and similar liquids. Figs. 43 and 44 show these pumps arranged with open impellers. The standard enclosed impeller pump casing is used. Stationary

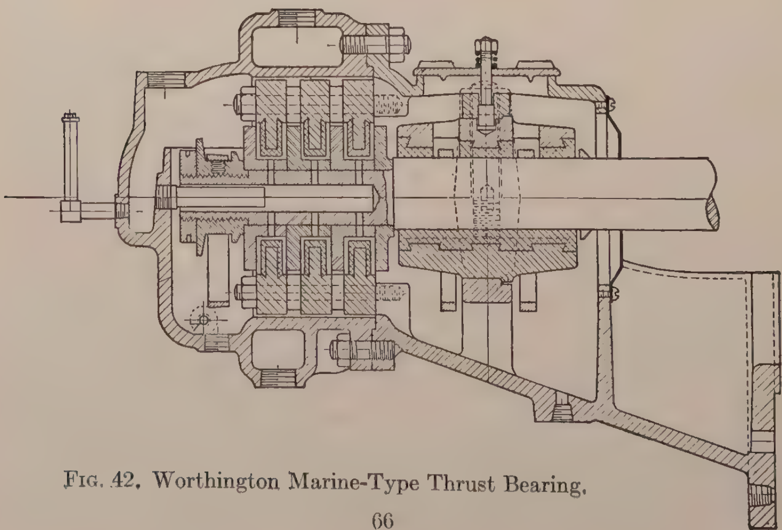


Fig. 42. Worthington Marine-Type Thrust Bearing.

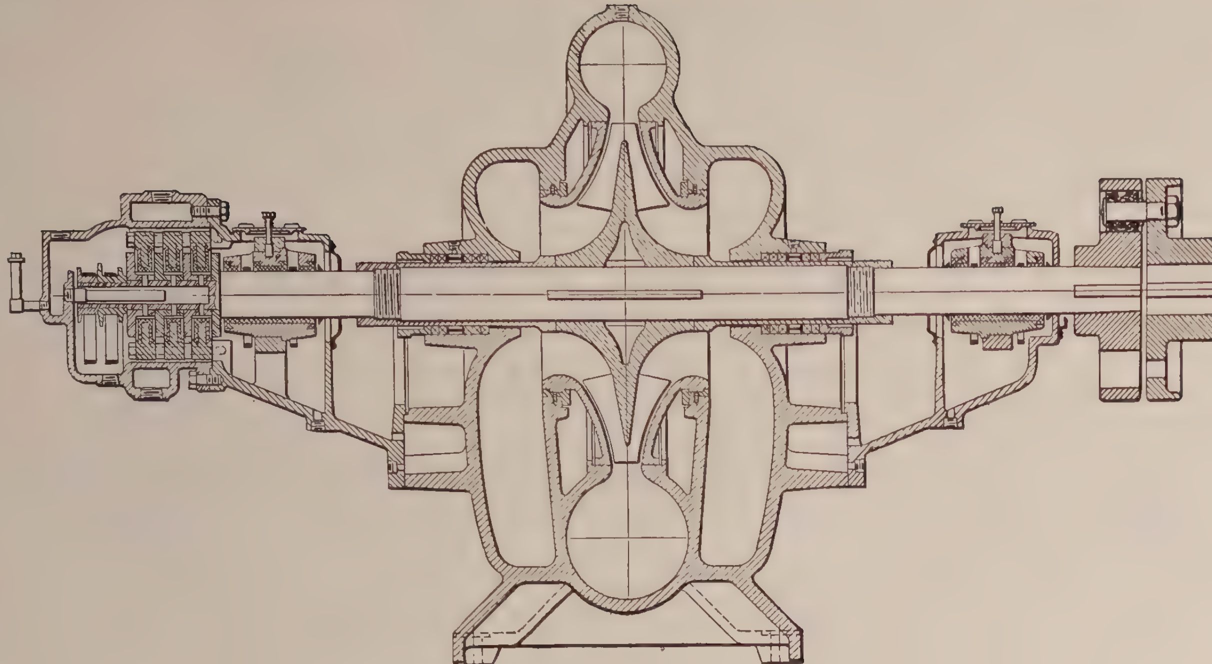


Fig. 43.

Sectional View of Open-impeller Volute Pump

For high-head service.

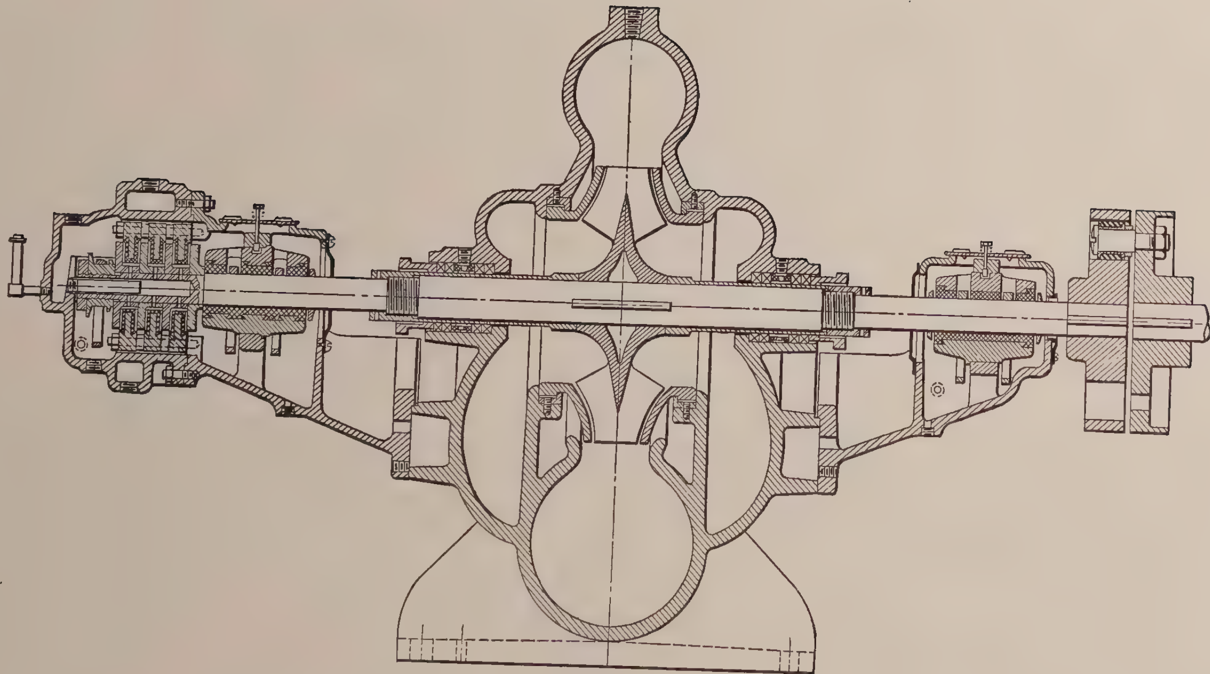


Fig. 44.

Sectional View of Special Open-impeller Volute Pump.

For high-head service

side plates are used to enclose the rotating impeller vanes. The side plates form a sealing joint over the entire surface of the vanes and retain the liquid within the vanes, and give a graduated pressure difference from the inlet to the outlet. The impellers are designed with the minimum number of vanes so as to give the greatest area possible through the impeller.

174. It is impossible to operate these pumps without a good **thrust bearing** as the nature of the liquid handled prevents an hydraulically balanced condition. A marine type of thrust bearing, Fig. 42, is preferred as it will prevent the impeller from being forced out of center by material becoming wedged between the impeller vanes and side plates.

175. The **open impeller** pump is specially designed for the pumping of fibrous or pulpy mixtures, sewage or other liquids containing solids in suspension. When pumping magazine paper stock, binding wire and strings wind around the revolving shaft sleeves and accumulate a mass of material that will eventually clog up the entrance to the impeller. To prevent this, Worthington pumps are fitted with a stationary protecting sleeve, and a special device effectively prevents the formation of such a mass.

176. Any material that lodges between the side plates and the impeller is quickly cut out by the sharp edges of the impeller vanes rotating at high speed.

177. Class P Pulp Pump.—For unusually thick liquids the Worthington Class P Pulp Pump, Fig. 45, has been developed, with end-type single-suction open impeller. The side wall of the casing forms the seal for the impeller which is designed with the minimum number of vanes of the maximum width.

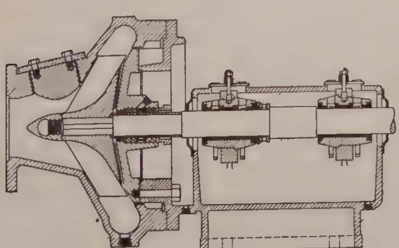


FIG. 45. Cross section of Worthington Class P Pulp Pump for unusually thick liquids.

178. The open-impeller pump is naturally not as efficient as the closed-impeller type, but it is more effective in handling pulpy or fibrous liquids. This type of pump will handle

any liquid of this nature that it is possible to make flow in a pipe line.

179. Volute Pumps for Special Service.—How alterations to existing patterns will fit standard pumps for special service is illustrated by connecting two volute pumps in series. The alterations necessary are usually confined to the bedplate and the series piping.

179A. In water-works service where the capacity is five million gal. per 24 hours and above, it is standard practice to connect in series two pumps of the volute type to obtain heads in excess of 200 ft. The most efficient form of unit is obtained in this way. To prevent an unsightly appearance and to simplify the series connection, a pump with bottom suction is used and the series pipe located under the floor. Pumps in capacities up to 50 million gal. per day have been built in this way.

180. Fig. 46 illustrates two single-stage volute pumps in series for water works service. These pumps deliver 50 million gal. per day at 280-ft. head.

181. Multi-Stage Volutes, Single-Suction.—There are several pumps of the multi-stage, single-suction type on the market today

Fig. 46. Two Worthington Single-stage Volute Pumps connected in series for water works service. Steam-turbine drive. Capacity 50,000,000 gal. per 24 hours at 280-ft. head.

in which the manufacturers have not made any attempt to properly handle the water from one stage to another. This results in excessive end thrust which these manufacturers have endeavored to counterbalance by an hydraulic thrust disk, with indifferent success. The hydraulic thrust disk, however, is successful, providing the pump is designed with proper staging effect for the water.

182. The unique way in which Worthington has applied the **volute principle for staging** purposes to the single-suction multistage pump is shown by Fig. 47. The liquid passages are full volute which expand both axially and radially, and continue in a volute path to form the suction chambers of succeeding stages. The separating walls of the stages are cast integral and act as a series of internal ribs, giving a casing construction of unusual strength. This pump is designed for 600 lb. per sq. in. discharge pressure.

183. The hydraulic balancing disk is employed to take care of any end thrust. The balancing disk is designed with a long leakage joint and a hub diameter nearly equal to the diameter of the impeller eye. The effect is to obtain a balance plug, which, when being acted upon by the discharge pressure, counterbalances 80 per cent thrust of the impeller directly and without depending upon the thrust pressure first passing through a leakage joint. The remaining unbalance will be taken up by the pressure supply from the long leakage joint formed by the hub of the disk. Large radial wearing surfaces restrict the amount of leakage between faces and work on a small pressure per square inch of surface as they have only 20 per cent of the pump thrust to take care of. The bearing disk together with the stationary wearing surfaces are of steel forgings. This material has been found to be superior to bronze where high pressures are involved as there is practically no erosion and the water passing between the surfaces when they are in rotation gives them a high polish.

184. Pumps for High Pressures.—Worthington has also developed a special eight-stage pump for 850 lb. per sq. in. discharge pressure. The casing of this pump is good for 1350 lb. per sq. in. discharge pressure and represents the extreme to which a special centrifugal

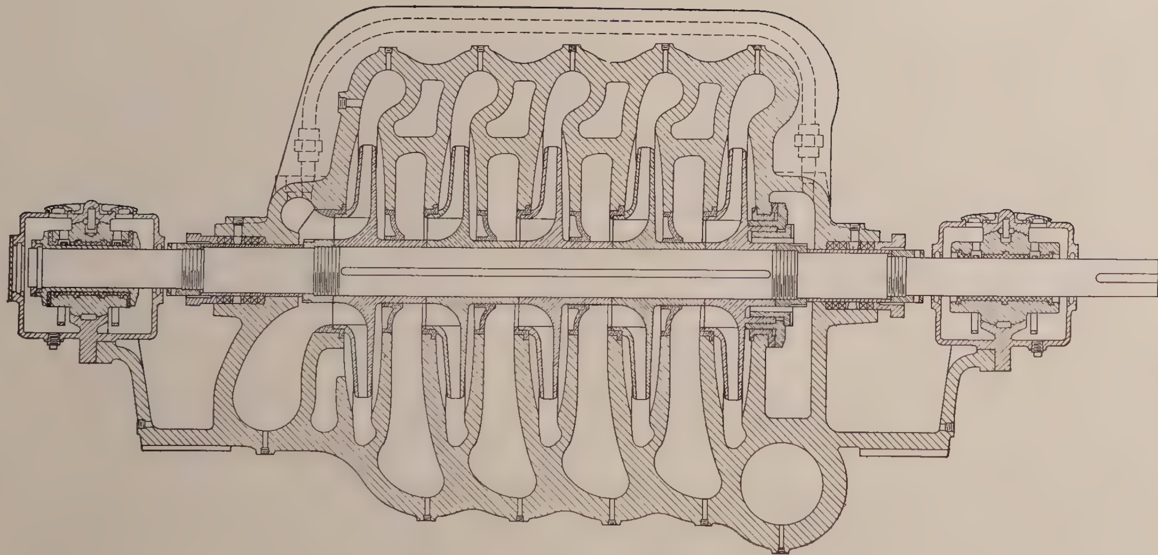


Fig. 47.

Cross Section of Worthington Multi-stage Single-suction Impeller Pump for High Heads.

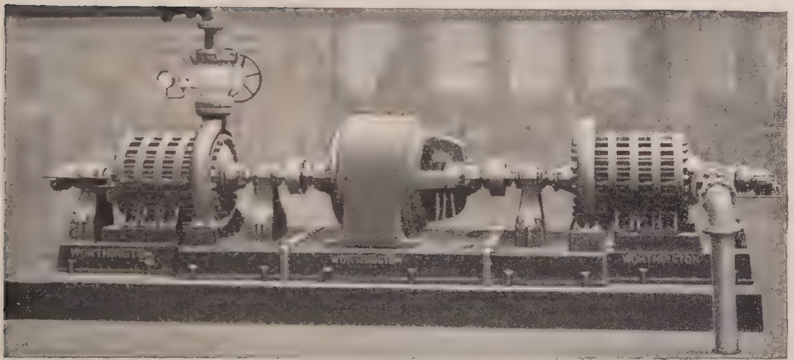


FIG. 48. Worthington 10-stage Mine Pump, pumping 1200 gal. per min. against a head of 1385 ft.

pump may be built to withstand high pressure and maintain the desirable feature of a split casing. Several installations have been made in mines of 8, 10 and 12-stage pumps for heads of 1000 to 1800 ft., each representing a special development for a special service. Fig. 48 is a typical 10-stage mine pump for 1200 gal. per min. against a total head of 1385 feet.

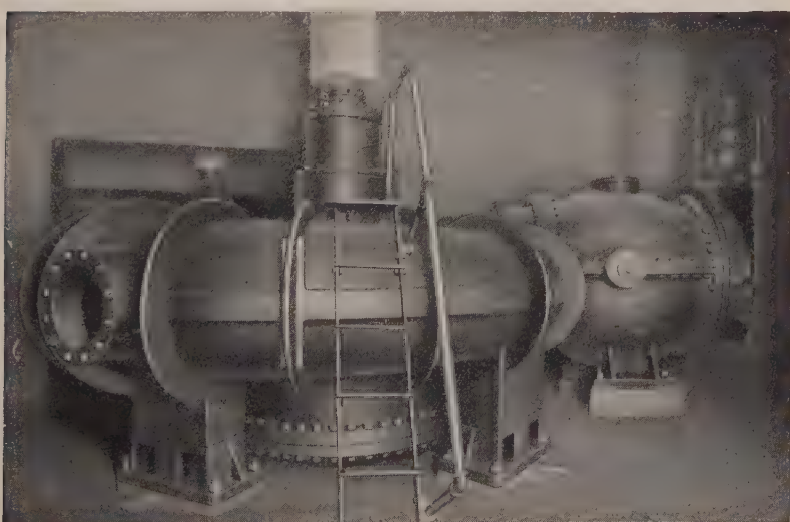


FIG. 49. Typical installation of Worthington Vertical Bottom-suction Centrifugal Pump.

185. Vertical Pumps.—With the exception of the sump pump described in Par. 49, all vertical pumps are regarded as special, as each installation presents a problem in itself. Worthington vertical pumps are available for pumping out dry docks, drainage and sewage-disposal work, etc., where a pump of the vertical type is necessary. Fig. 49 and Fig. 50 show typical installations of vertical pumps built by Worthington.

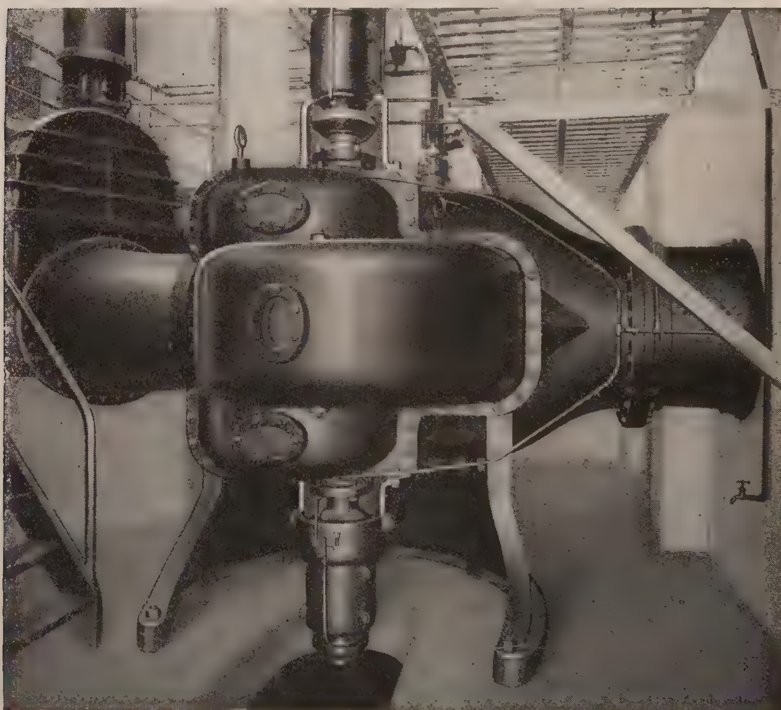


Fig. 50. Typical installation of Worthington Vertical Double-suction Volute Pump.

SECTION III

Power Pumps

(Figures refer to paragraph numbers)

Types, drives, construction, 1–69; Speeds, capacities and pumping cost data, 75–80; Vertical triplex pumps, single-acting, sizes and data, 85–93; Vertical triplex pumps, single-acting, forged steel liquid ends, 94-98; Stuff pump, sizes aud data, Triplex pump with automatic receiver, sizes and data, 100; Vertical triplex power pumps, double-acting, sizes and data, 101; Horizontal single power pumps, sizes and data, 103-106; Horizontal duplex power pumps, sizes and data, 106-111; Horizontal enclosed, triplex power pumps, sizes and data, 112-114; Horizontal enclosed, duplex power pumps, sizes and data, 115-117.

SECTION III

Power Pumps

1. In pump nomenclature, the term "power pump" has but one accepted meaning, and is only applied to positive-displacement pumps operated through cranks and connecting rods, by the application of power to the crank-shaft. A power pump works just like a reciprocating engine, except that the power action is reversed. In the engine a piston causes a shaft to rotate through the action of a crosshead, wrist pin, connecting rod, crank-pin and crank. In a power pump, external power is applied to the crank-shaft causing it to rotate, and by the same chain of parts as in an engine, a piston or plunger is caused to reciprocate.

2. A power pump differs essentially **from a steam pump** in many of its operating characteristics. These differences should be clearly understood, as often a power pump replacing a steam pump does not give complete satisfaction until the user becomes educated to the requirements for satisfactory power-pump operation.

3. A power pump is essentially a constant-speed machine. It does not have the flexibility as regards speed and capacity that a steam pump has. Therefore, more care must be taken to insure that the **power-pump speed** is correct for all the conditions to be met. If the suction pipe is too long or too small, or if a full supply of water is not available, a steam pump can readily be slowed down until its capacity is equal to the maximum amount of water which can be drafted through the suction line. A power pump under similar conditions must run along at full speed, but only partially filling. This causes severe pounding and, eventually, breakage.

4. If **excess pressure** is put on the discharge line, as may happen from a closed or partially closed valve, the steam pump automatically slows down and often stalls if the pressure becomes high enough. Under the same conditions the power pump goes right along at full speed until the motor burns out under overload, or until something breaks on the pump. It is, therefore, essential that a power pump be protected by a spring **relief valve** to prevent the discharge pressure becoming too great.

5. The above considerations also make it desirable that a power

pump be fitted with liberal air chambers on both suction and discharge.

6. Power pumps may be of the vertical or horizontal types, with inside-packed pistons or outside-packed plungers.

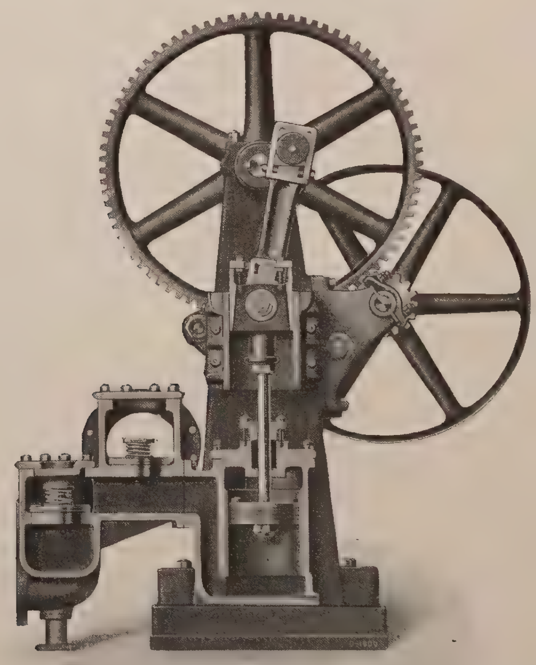


Fig. 1. Typical section of Worthington Vertical Power Pump. Double-acting, packed-piston type.

7. Fig. 1 is a typical section of a vertical double-acting **inside-packed-piston** pump. A horizontal pump of the double-acting inside-packed piston type is shown sectionally by Fig. 2.

8. A double-acting pump will have about twice the capacity of a single-acting pump with the same diameters and stroke. Where the liquid to be handled is clear and the pressures are comparatively low the double-acting **packed-piston pump** works well and has the **advantages** of smaller stuffing boxes, and larger capacity for a given size.

9. A vertical single-acting **outside-packed-plunger pump** is shown sectionally by Fig. 3. In the single-acting pump the plunger pack-

ing is outside, visible and easily adjusted to prevent leaks. The plunger pump is more suitable for high pressures or gritty liquids, since leaks due to pressure or wear are at once apparent and more easily corrected than would be the case of leakage past the packing of a piston, inside a double-acting pump.

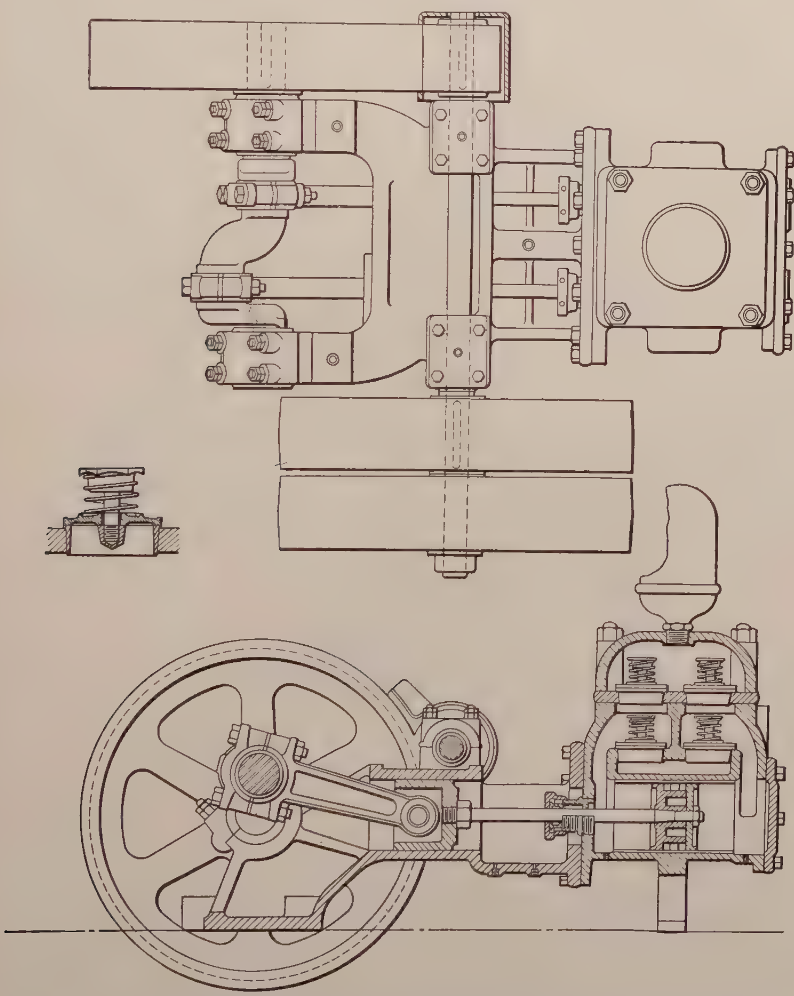


Fig. 2. Typical section of Worthington Horizontal Power Pump. Double-acting, packed-piston type.

10. Classification of Power Pumps.—Power pumps are classified, as to power end, into belt-driven or direct-connected, and according to the number of cranks, into single (1 crank); duplex (2 cranks); triplex (3 cranks).

11. The method of driving any power pump can be varied to suit the conditions prescribed by the purchaser. In hotels, office buildings, apartment houses, etc., where noise is objectionable, the belt

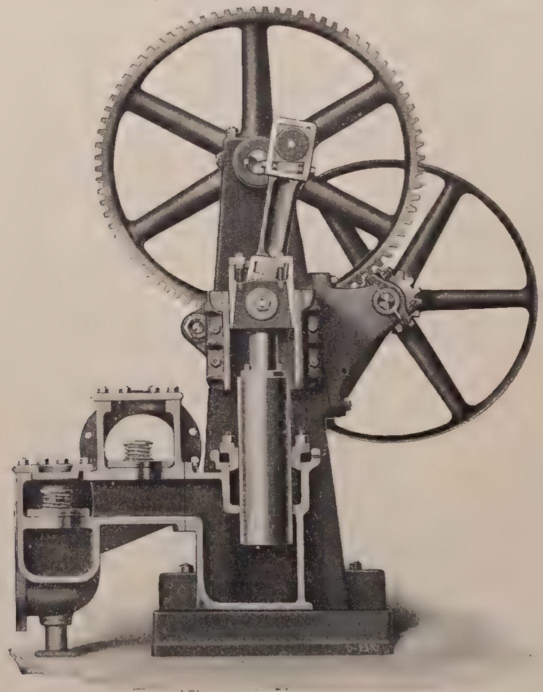


Fig. 3. Typical section of Worthington Vertical Power Pump. Single-acting plunger type.

drive is recommended. In mills, factories and mines, where noise is not objectionable, the electric gear drive is used.

12. Belt-Driven Pumps.—Where a power pump can be located near a line shaft, the easiest way is the **belt drive**. In some cases, where the speeds of the pump and shaft are suitable, the driving pulley can be placed on the overhung end of the crank-shaft. With the usual line shaft speeds, it is considered good practice to equip power pumps with an auxiliary or pinion shaft on which tight and

loose pulleys are located. This permits the pump to be started and stopped by belt shifting. The pinion shaft carries a pinion which meshes with a large gear on the crank-shaft giving the necessary reduction in speed. By selecting the proper ratios for the gears and pulleys used, the correct speed for the pump can be had from practically any line shaft running at the ordinary speeds in use today.

13. Power pumps may also be driven by a belt from an internal combustion engine, electric motor or other prime mover. Fig. 4 illustrates a power pump belt driven from an electric motor. The prime mover is located sufficiently far from the pump to insure a liberal arc of belt contact on both driving and driven pulleys. Fig. 5 shows a pump driven from an electric motor by a short belt. An **idler pulley** is used on the slack side of the belt to increase the arc of contact on the driving and driven pulley. This arrangement is very compact and is used extensively.

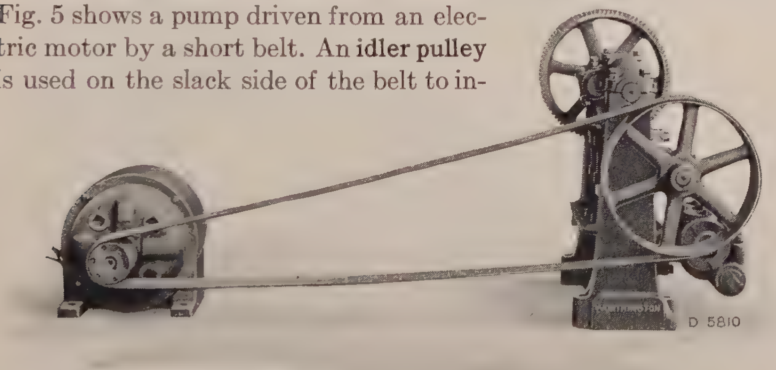


FIG. 4. Worthington Vertical Power Pump, Belt-driven.

14. The belt drive operates with the minimum of noise. Excess pressure on the liquid end of the pump will throw the belt or cause it to slip, preventing any damage to the pump.

15. Belt drives are simple, flexible, inexpensive, and especially when **compared to double-reduction gear** drives, the belt drive is quiet and easy to keep in order. The double-reduction gear drive gives noisy operation and with high-speed gearing its operation is extremely noisy. The double-reduction gear drive is less efficient than the belt drive. The gears, due to the high rim speed, are subject to rapid wear. Moreover, all shocks are transmitted directly from the pump through the gears to the motor.

16. With the belt drive, a higher speed motor can usually be used than with double-reduction gears. Hence, the **belt-driven outfit is cheaper.** The belt-driven machine is cheaper since the motor subbase and motor gears are not required for the belt drive. Moreover, with large motors, double-reduction gear drive requires an outboard bearing which in its turn increases the cost of the motor.

17. Belted pumps always get **quicker shipment** than geared pumps. Stock pumps are always fitted for belt drive, whereas for pumps fitted with double-reduction gears it is necessary to wait for motor

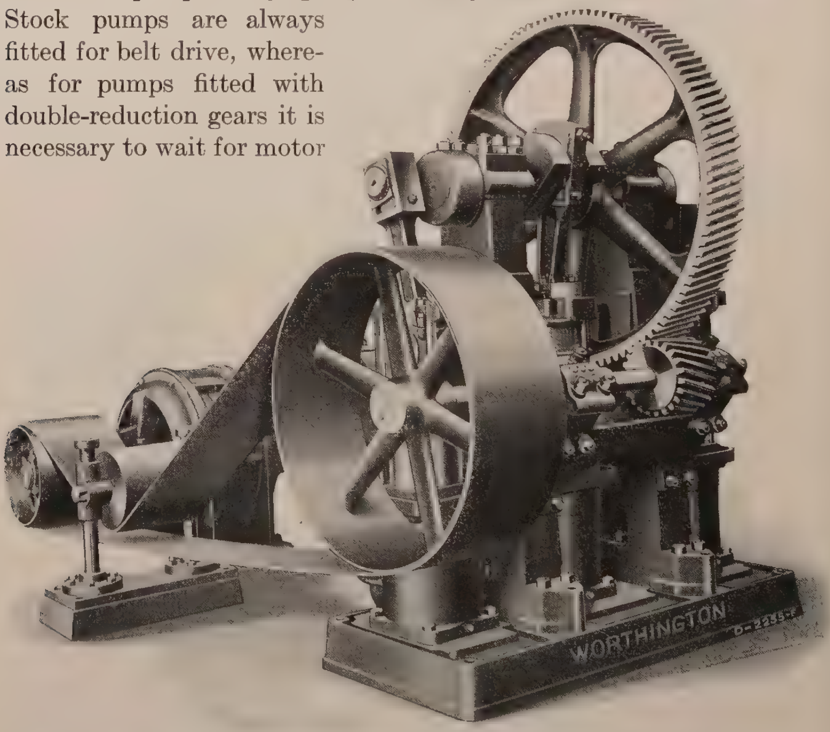


Fig. 5. Worthington Vertical Triplex Power Pump driven by short belt.

prints before motor gears and motor subbase can be laid out, and usually there is delay in getting the motor to the factory in order to fit the motor to the pump.

18. Change of speed is easily accomplished on a **belt-driven machine** by simply changing belt pulley, whereas the change of speed with a double-reduction gear pump is expensive as it requires a new set of motor gears and the resetting of the motor.

19. This change of **speed** is a very important factor to remember in reference to **deep-well pumps** where the water level is very apt to fluctuate and the water supply to vary.

20. With **well heads** where the head is to be moved back from over the well for the **inspection** of the working rods and the working barrel, the double-reduction gear drive makes the shifting back and forth a rather difficult and slow process. Especially is this true if the motor is fairly large and is fitted with an outboard bearing. When the well head is returned to its original position it must be relocated in perfect alignment with the motor. On the

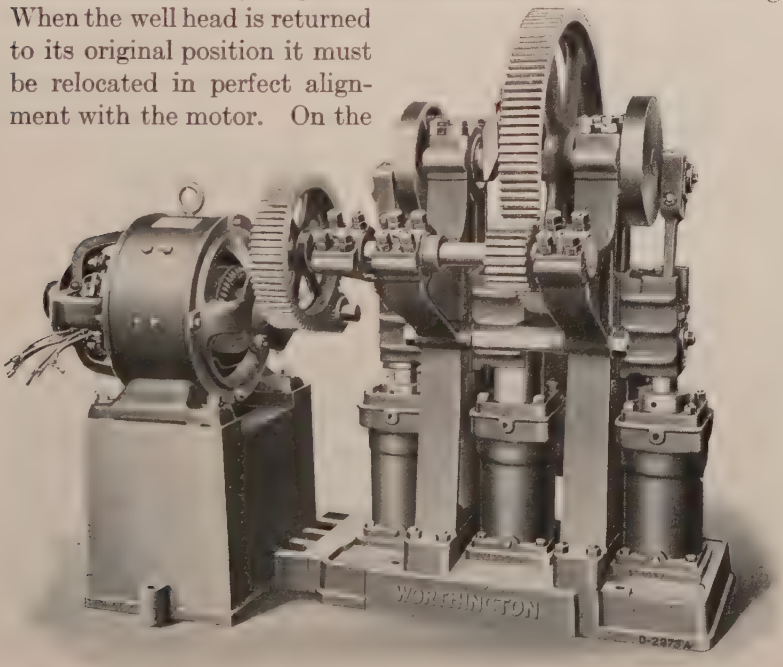


Fig. 6. Worthington Vertical Triplex Power Pump, single-acting plungers, motor-driven, with double reduction gears.

other hand, with belt drive all that is necessary before the head is moved is to throw the belt off, and when head is put back in place the belt can be readily put back on the belt pulley.

21. Direct-Driven Pumps.—The direct gear-driven power pump is a very compact unit, which is its chief advantage. Fig. 6 shows a Worthington Power Pump arranged for direct drive by an electric motor through a double-reduction gear. The gears have cut teeth.

The motor pinion, and sometimes the main pinion, is made from rawhide, fibre or bakelite to reduce the noise. The motor is mounted on a subbase bolted to an extension of the pump base. Fig. 7 illustrates a Worthington power pump direct-connected to a Worthington **Diesel Engine.** This is a highly efficient pumping unit, as both engine and pump have the highest efficiency known for machines of their respective classes. Clutch couplings are provided between the pump and engine for starting. These sets are recommended for stand-by service and for locations where steam or electric power is not available.

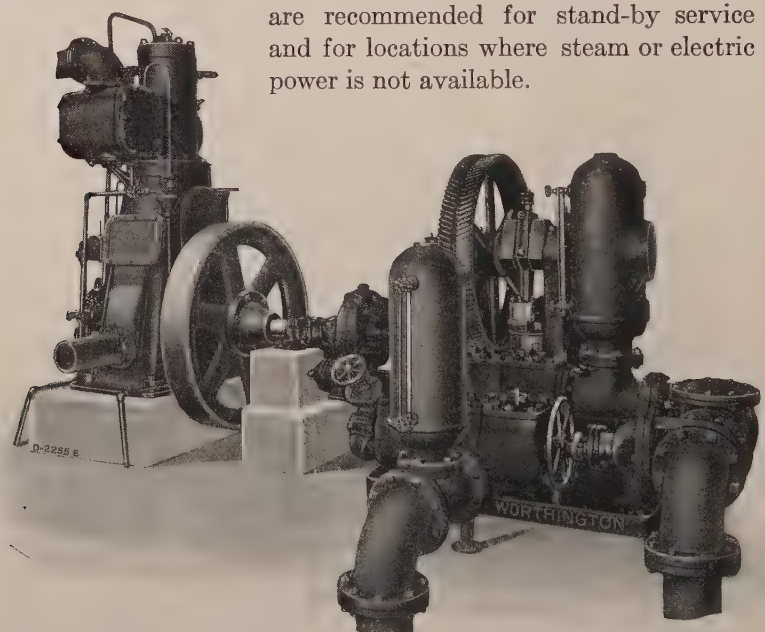


Fig. 7. Worthington Vertical Triplex Power Pump driven by Worthington Diesel Engine.

22. For the larger sizes of pumps direct-geared, the preferred drive is **single-reduction** herringbone **gears** and flexible coupling, as shown by Fig. 8. This arrangement is more compact and more efficient than double-reduction gears. With the single-reduction gears and flexible coupling, no outboard bearing is required on the motor, and this reduces the cost of the motor. The motor need not come to the factory for assembly with the pump and this saves extra freight on the motor. Furthermore, the pump does not need to await the receipt of motor so that quicker shipment can be made

of the pump. The arrangement with single-reduction gears is more accessible, both as regards pump and motor. No motor subbase is required as the motor can sit directly on the concrete foundation.

23. A **single power pump** has one crank which operates one double-acting piston (or plunger) or two single-acting plungers. Fig. 49 shows a single pump.

24. A **duplex power pump** has two double-acting pistons (or plungers) or four single-acting plungers operated by cranks 90 deg. apart. A duplex pump is shown by Fig. 52.

25. A **triplex power pump** has three pistons or plungers operated by cranks 120 deg. apart. They may be of the double-acting type as shown by Fig. 47 or they may be of the single-acting type as shown by Fig. 37.

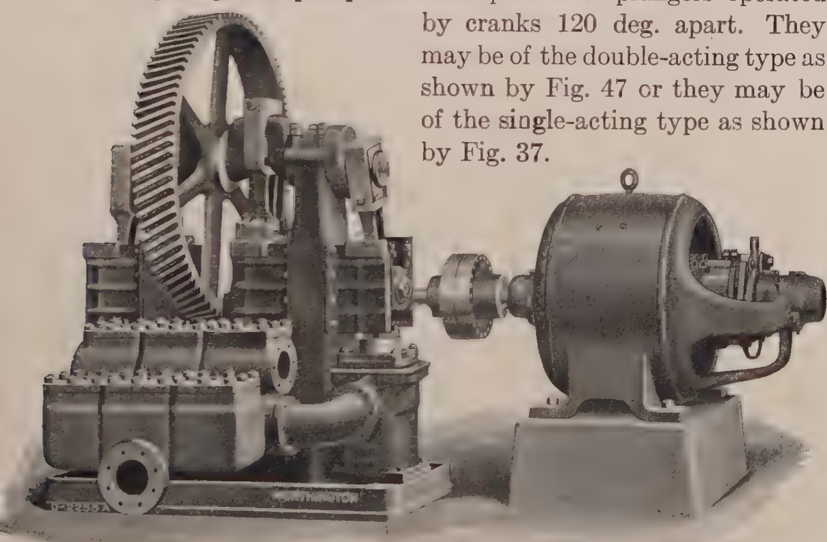
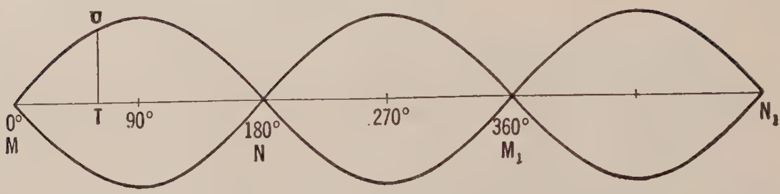


FIG. 8. Worthington Vertical Triplex Power Pump single-acting plungers, driven by electric motor through single reduction gears.

26. The comparative merits of the three classes may be fully understood by a study of the **flow curves** of the single pump double-acting, Fig. 9; the duplex pump double-acting, Fig. 10; the triplex double-acting, Fig. 11; and the triplex single-acting, Fig. 12.

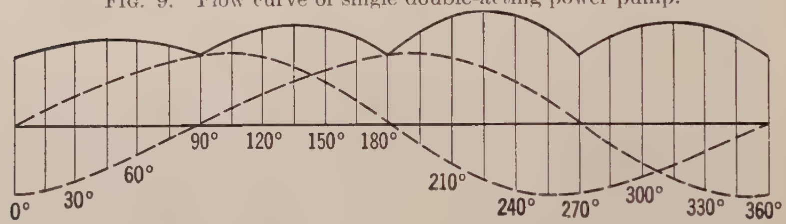
27. The rates of suction and discharge of the individual pistons or plungers are shown by broken lines and the resultant rate by heavy full lines. These **curves** are **plotted** for one revolution of the crank shaft.

28. Suppose first a single-plunger power pump, like Fig. 3. The



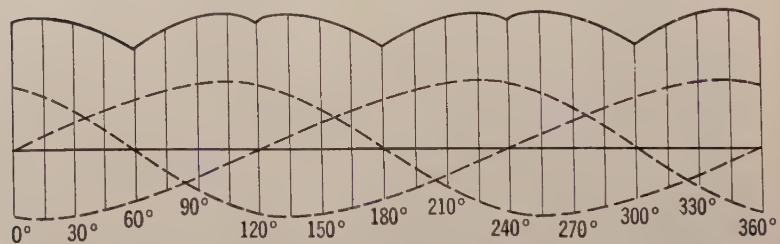
SINGLE DOUBLE-ACTING

Fig. 9. Flow curve of single double-acting power pump.



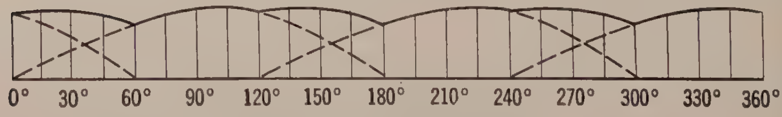
DUPLEX DOUBLE-ACTING

Fig. 10. Flow curve of duplex double-acting power pump.



TRIPLEX DOUBLE-ACTING

Fig. 11. Flow curve of triplex double-acting power pump.



TRIPLEX SINGLE-ACTING

Fig. 12. Flow curve of triplex single-acting power pump.

crank B (see Fig. 13) is rotating at a uniform velocity. This causes the plunger to have a variable velocity; and if we ignore the slight distortion due to the angularity of the connecting rod, the velocity of the plunger is at any instant equal to the velocity of the crank-pin B, times the sine of the angle which the pin has moved through from its dead center.

29. Expressed as a formula, let V_b be the linear velocity of the crank-pin, and V_c the linear velocity of the crosshead; and let α be the angle through which the crank has moved from the outer dead center. Reduced to diagrammatic form as in Fig. 13, it is clearer. B has moved from dead center M through angle α as shown. The linear velocity of B is shown by the tangent BE, to a definite scale of so many feet per minute or second = 1 inch. If we draw the horizontal BF and the vertical EF, we have resolved velocity V_b = BE into its horizontal and vertical components.

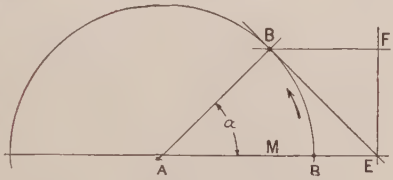


FIG. 13. Crank motion diagram.

30. The velocity of the wrist pin is practically the same as BF so we can call BF, or V_c, the velocity of the wrist-pin, which is also the velocity of the plunger at the instant the crank has moved through α angle from dead center M. In the right triangle BEF, by plane trigonometry, BF = BE × sine of angle BEF; or,

$$V_c = V_b \sin E$$

31. But the angle at E is the same as angle α since their sides are mutually perpendicular to each other, BE to AB, and EF to AM. We can therefore write

$$V_c = V_b \sin \alpha$$

32. If now we consider the circular path of the crank-pin "developed" into a straight line, we have Fig. 14. Point 0, where it starts, is dead center M. Let MM_1 be taken as one revolution or circumference. In the original layout, this was made 6 in., so that 1 in. = 60 deg. Take a point 1 at 60 deg. or 1 in. from M; then let the line TU be the velocity of the plunger at this position of the crank-pin. It will be equal in general to V_c. sin α; in this case, $V_b \times 0.866$, since 0.866 is the sine of 60 deg. Take for example a 4x6 triplex, running 50 r.p.m. This pump has 6-in. stroke, so the crank radius, AB, is 3 in. In one revolution the crank-pin will travel one circumference, or 18.85 in. Its speed in feet per second will be

$$\frac{18.85 \times 50}{60 \times 12} = 1.31.$$

At the position shown, then

$$V_c = V_b \sin \alpha = 1.31 \times 0.866$$
$$= 1.134 \text{ ft. per sec., or 68 ft. per min., approx.}$$

33. In the same manner as the point U was calculated, we can find enough other points between M and N to plot the curve MPN. From N to M_1 we get the same values, but we plot the curve below the line MM_1 instead of above it, as the plunger is moving in the opposite direction, and is on suction instead of delivery. Since the cross-section of the plunger is uniform, the flow of water into the delivery line is directly proportional to the velocity V_c.

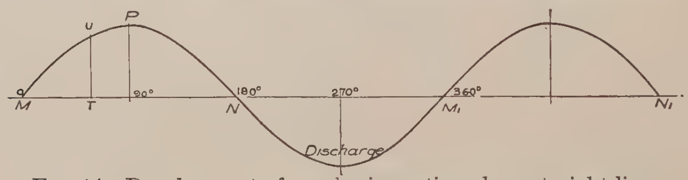


FIG. 14. Development of crank-pin motion along straight line.

If then we had a single plunger power pump, water would start to flow into the delivery line at M, and would continue to flow faster until the maximum speed was reached at P; it would then start to flow more slowly until N, when it would stop, and the plunger would then start on suction, and fill the pump chamber from N to M_1, starting at M_1 again to flow into the delivery line. It can thus be seen that the flow from one single-acting plunger is subject to considerable fluctuation in both suction and delivery lines. This led to the addition of two more plungers with their connecting rods, cranks, shaft, etc., setting all three cranks 120 deg. apart.

34. In Fig. 15 let A be the crank-shaft; B_1 is one crank, B_2 is the second crank and B_3 the third. As the crank-shaft revolves in counter-clockwise direction, the plungers working vertically, crank B_1 will cause the action represented by curve 1 in Fig. 16. Crank B_2 being 120 deg. behind crank B_1, will have the effect shown by curve 2 and crank B_3 the action shown by curve 3. Starting from M on curve 1, we see that by the time the discharge curve reaches point c, over P on curve 2, crank B_2 has begun to cause a discharge which increases from P to d while

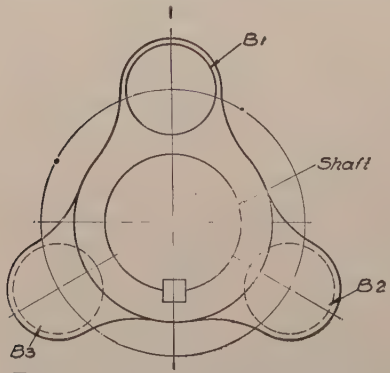


FIG. 15. Location of crank pins on triplex pump shaft.

curve 1 decreases from c to N. So both plungers are delivering while shaft rotates the 60 deg. from P to N, and the true delivery curve between these points is a curve cd, the ordinates of which are the sum of the ordinates of cN and Pd. In like manner, at R the discharge due to crank B_3 commences as that due to B_2 is falling off, and the true discharge curve is ef. Also, though we started with the action of crank 1 at M, the true discharge here was being helped out by crank 3, so if the pump is started from rest at M, the net or true discharge curve really begins at point *a*.

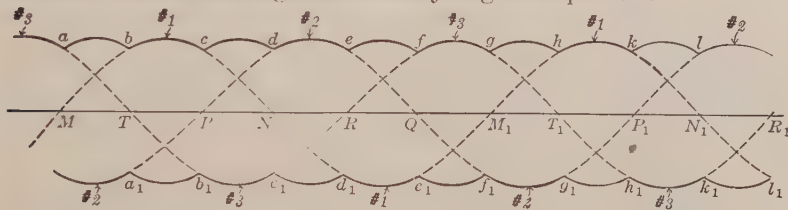


FIG. 16. Combined crank motions of triplex pump which correspond with the flow of water.

35. Thus the **delivery of a triplex single-acting plunger pump** is the composite curve *abcd . . . . l*, etc.; and the suction curve is $a_1 b_1 c_1 d_1 \ldots l_1$, both practically alike, and both approximating to a straight line. Owing to the inertia of moving water, if a recording gage were placed on the delivery line of a well designed triplex pump, the curve drawn by it would be even smoother shown in Fig. 16.

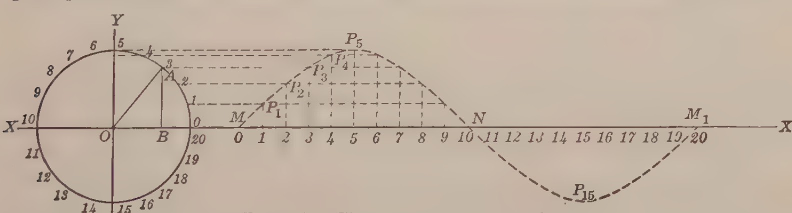


FIG. 17. Sine-curve construction.

To construct a "sine-curve," draw two axes at right angles, *XX* and *YY*, crossing at *O*. With *O* as a center, draw a circle with radius *OA*, equal to the highest ordinate of the curve. Start at zero and lay off any desired number of equal parts on circle (five to quadrant is convenient). Start at point *M* on the line *XX* and lay off MM_1 equal to the circumference of the circle. Divide MM_1 into the same number of equal parts as were laid off on the circle. Number the points on both the circle and on MM_1. Locate points P_1, P_2, P_3, etc. by erecting perpendiculars at points 1, 2, 3, etc. on MM_1, and running horizontals from the corresponding points 1, 2, 3, etc. on the circle. Draw curve through points P_1, P_2, P_3, etc., obtaining curve M, P_5, N, P_{15}, M_1. This is one cycle or unit. If more cycles are desired, start at M_1, and repeat the curve already drawn as many times as may be necessary.

40. Vertical Triplex Pump.—The vertical triplex pump is probably the most used type of power pump. A brief description of the construction of the power and liquid ends will be of interest.

41. Pumps with a 6-in. stroke and some of the light pressure 8-in. stroke designs have "I" section **frames.** In these smaller pumps the "I" frame is quite satisfactory, but in the larger sizes, where stresses are greater, the uprights are box sectioned, as shown in Fig. 18. This is the stiffest design known.

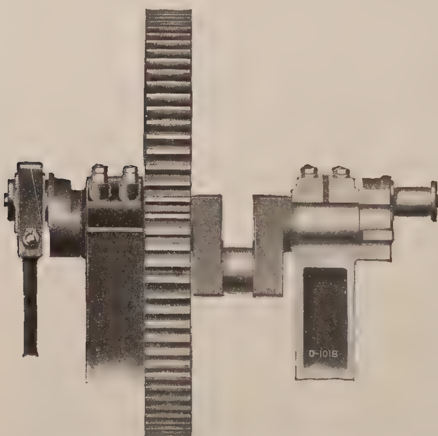


Fig. 18. Showing arrangement of gear, crank-shaft and frame uprights in Worthington Triplex Pump.

42. Large pumps have uprights, base and pinion-shaft brackets as separate parts bolted together, but in the smaller pumps the brackets are cast integral with the main frame. The one-piece construction is preferred in small pumps because of its simplicity and the certainty of stiffness. In the larger pumps, the advantages of sectionalized design are secured, together with greater ease of obtaining rigid connections.

43. Bearings are made of babbitt, scraped to a true fit with the shaft. The dimensions are such that wear is reduced and the chance of heating, even under maximum loads, is reduced to a minimum.

44. Standard **crank-shafts** are of the composite design, the shaft itself being offset to form the center crank, and cranks with pins are forced on each end of the crank-shaft to form the end cranks. Besides the security offered by a hydraulic press fit, the cranks are keyed to the shafts for additional rigidity. Crank-pins on the larger sizes are cast integral with cast steel cranks turned and ground, true and smooth. The one-piece crank and pin design makes this part extra strong, as there is no tendency for the pin to work loose in the crank with the heavy pressures thrown upon it every time the plunger is driven down.

45. A glance at Fig. 19 reveals an unusually sturdy construction that is typical of all Worthington Power-Pump crankshafts. Note the large diameter of the shaft, the heavy cheek pieces of the center crank, and the substantial design of the end cranks and pins. These liberal dimensions are largely responsible for the smooth action and the long **life** of Worthington Triplex Pumps.

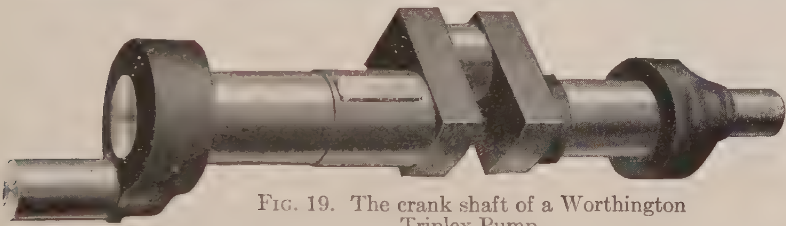


Fig. 19. The crank shaft of a Worthington Triplex Pump.

46. Fig. 19 shows the crank-shaft of a triplex pump. The crank-pins are set at an angle of 120 deg. apart so the plunger strokes overlap each other, giving not only a smooth flow to the discharge, but reducing to a minimum sudden strains upon the driving mechanism.

47. The **pinion shaft** is steel, finished all over to a polish. It is true and to exact size, so that it runs without vibration or heating in the journals. Both the pinion and tight pulley are securely keyed to this shaft.

48. All **gears** are made from a close grained special iron mixture of high tensile strength. Teeth are involute in form and accurately machined from solid blanks. The involute tooth is used because it allows a large amount of adjustment and preserves the teeth when bearings have become worn.

49. The location of the main gear is particularly well chosen, it being placed between the crank-shaft bearings, as shown in Fig. 18. The rigid support resulting from this arrangeent keeps the gears in true alignment.

50. All pumps with a stroke of 12 in. or less are fitted with tight pulley only.

51. Tight pulleys are always placed next to the pinion-shaft bearing, so that the heaviest belt strains are as close to the bearing support as possible. Loose pulleys are held in place by a collar on the outboard shaft.

FIG. 20. Single helical gears cut in Worthington shops for Worthington Power Pumps.

52. Outside **connecting rods** are of the solid end type with wedge and screw adjustable boxes at each end. (See Fig. 21.) The center rod is necessarily of the fork end type at its upper end.

53. Connecting rod boxes are lined with a high quality of bearing babbitt, scraped to fit crank and crosshead pins.

54. The **crossheads** are of the engine type with adjustable shoes working in bored guides. They have ample surface so that friction is reduced to a minimum. Shoes are made of cast iron, and since they work in guides of the same material, one of the best bearing combinations known is obtained. The arrangement provides adjustment for the crosshead to retain it in the center of the guide, so that the load is always uniformly distributed over the shoes. Fig. 22 shows the construction clearly.

55. The **crosshead guide** is separate from the frame and secured to it by studs and nuts. The guide has a circular bore and the crosshead shoes are turned to the same diameter so that any uneven wear in the guides does not throw the crosshead out of line, but simply causes the crosshead to turn slightly and adjust itself to the connecting rod and plunger.

FIG. 20a. Straight spur gears cut in Worthington shops for Worthington Power Pumps.

56. In the larger sized pumps **liquid cylinders** are separate castings secured to the base of the pump. Because of this construction replacements are easy and economical. Substitution of some other metal can easily be made should operating conditions demand that the cylinders be made of bronze or other special composition.

57. Stuffing boxes are extra deep and fitted with cast iron stud glands.

58. The smaller pumps usually have **one-piece**

cylinder and base construction because of the simplicity gained through eliminating extra joints.

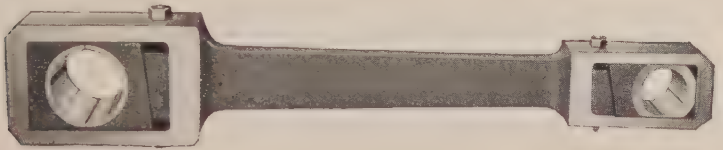


Fig. 21. Connecting-rod construction of Worthington Power Pumps.

59. The general arrangement of **plunger** in single-acting cylinder is displayed in Fig. 23. Plungers are made of special hard cast iron, ground to size.

The method of securing plungers to the crossheads in the large size single-acting pumps is shown in Figs. 22 and 23 which also show at a glance how easily the plunger may be disconnected and removed. The arrangement also allows a plunger of special composition to be used in a standard pump to meet the needs of special requirements, such as handling chemicals, acid-water, etc.

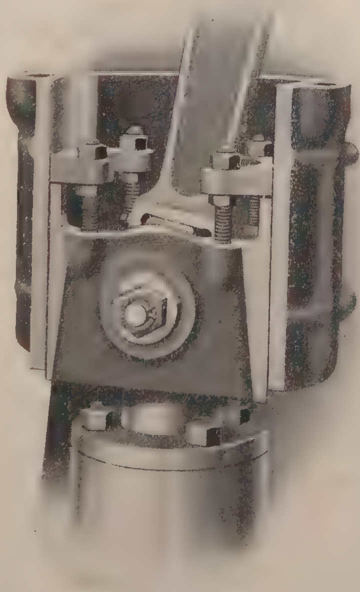


Fig. 22. Cross-head construction of Worthington Power Pumps.

60. In double-acting pumps the **pistons** are ordinarily made of gray cast iron, but other metals may be substituted for special conditions. They work in brass-lined cylinders like that shown in Fig. 24, and pistons are packed with several rings of fibrous packing held in place by followers. A nut, lock nut and cotter pin secure each piston to its rod and prevent the piston from backing off. The piston packing is accessible through the top of the cylinder by raising the stuffing-box head and piston follower. The rear of the pump is clear for access to the packing.

61. The design of **valve chests** varies over a wide range to suit the capacities and pressures for which the various pumps are designed. The external view of each pump must serve as an illustration of valve-chest design in each particular case, as no detail view can be truly typical for every pump. There is one characteristic, however, common to all Worthington valve chests; that is, the provision for

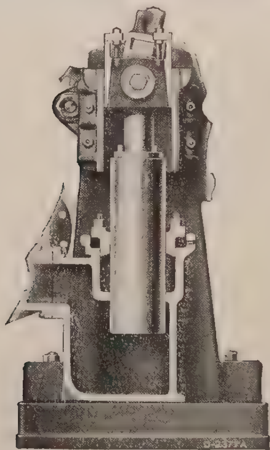


Fig. 23. Single-acting Plunger of Worthington Vertical Power Pump.

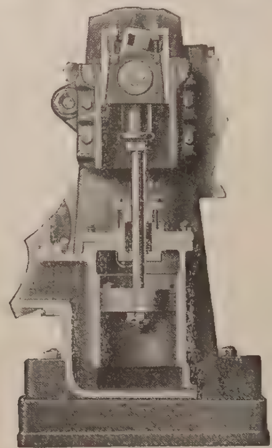


Fig. 24. Double-acting piston of Worthington Vertical Power Pump.

easy access to the liquid valves. Valves are accessible from above, seats can be refaced easily; all chests have these features.

62. Fig. 25 shows the construction of a valve chest used in the Worthington Triplex Pump. The details of design vary somewhat in the several types of pumps, but in general, the comments given herewith on the placing of the valves and their accessibility apply to all Worthington Triplex Pumps.

63. Each valve, or set of valves for larger pumps, is in a separate compartment. A handhole plate is bolted to the chest above each valve compartment so that access to the valve is made extremely easy. The compartments are large and there are no out-of-the-way pockets from which the valves must be extracted in order to remove them for inspection. The valves can be reached directly and simply.

64. Attention is called to the sturdy construction, the unusual thickness of metal used and the simplicity of the design in general.

65. Rubber disk valves with bronze seats, stems and springs are ordinarily supplied with Worthington Triplex Pumps. The valves are made of either hard, medium or soft rubber, according to the temperature of the water and the pressure pumped against.

66. Bronze disk valves can be furnished at no extra charge should this type be preferred by the customer.

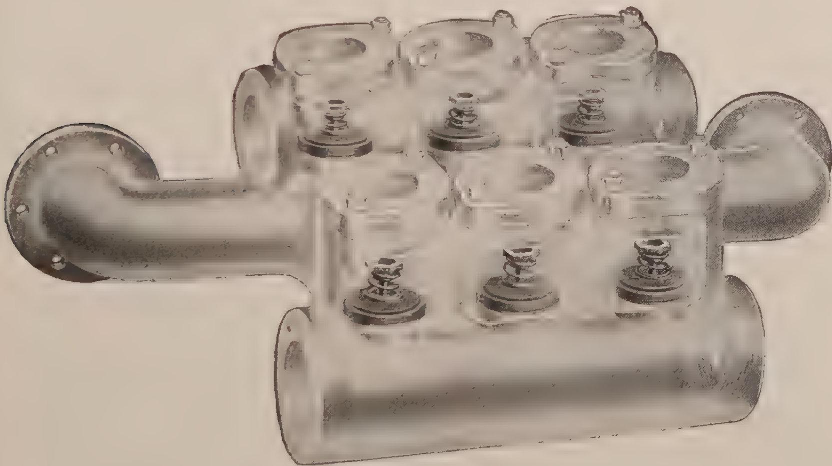


Fig. 25. Valve chest of Worthington Triplex Power Pumps.

67. When tar, heavy oils, molasses, paper stock and similar viscous or fibrous substances are handled, the pump can be fitted with **ball valves.** In some designs the ball valve is standard construction, and other pumps that have fluid passages suitable for handling thick liquids can be fitted with ball valves.

68. For high pressures various types of **wing valves** are used.

69. In section V, devoted to direct-acting steam pumps, and in section VII, on oil pumping, different types of liquid ends, pistons, plungers and valve service are discussed. Any type of liquid end used with direct-acting steam pumps can also be fitted with a power end. The Worthington Pump and Machinery Corporation has patterns and drawings for an extensive line of such pumps. For special service of any kind, power pumps can be designed and built to suit practically any conditions of pressure, capacity, or power desired or available.

75. Speeds and Capacities of Power Pumps.—The common forms

of standard power pumps are three: Triplex single-acting; duplex double-acting, and triplex double-acting. Their speed-capacity relations are easily calculated from the following equations:

Triplex single-acting

$$\text{r.p.m.} = \frac{102 \times \text{gal. per min.}}{\text{Stroke} \times \text{Dia.}^2} = \frac{\text{gal. per min.} \times 80}{\text{Plgr. Area} \times \text{Stroke}}$$

Duplex double-acting

$$\text{r.p.m.} = \frac{76.5 \times \text{gal. per min.}}{\text{Stroke} \times \text{Dia.}^2} = \frac{\text{gal. per min.} \times 60}{\text{Plgr. Area} \times \text{Stroke}}$$

Triplex double-acting

$$\text{r.p.m.} = \frac{51 \times \text{gal. per min.}}{\text{Stroke} \times \text{Dia.}^2} = \frac{\text{gal. per min.} \times 40}{\text{Plgr. Area} \times \text{Stroke}}$$

If instead of revolutions per minute, the speed is desired in feet per minute, substitute **6** for **stroke**, and the answer will be feet per minute. The above formulas all include an allowance of 4 per cent for slip.

76. Cost of Pumping.—The following equations or formulas give the cost of pumping 1000 gallons of water with various types of power:

77. With **Motor Drive** the cost of electricity in cents per 1000 gallons of water is

$$\frac{h \times k}{e \times m \times 319}$$

$$\frac{p \times k}{e \times m \times 138}$$

where h is the total head in feet;

p the pressure in pounds per square inch;
k the cents per kilowatt-hour;
e the pump efficiency, and
m the motor efficiency.

78. With **Oil Engine Drive** the cost of oil in cents per 1000 gallons of water is

$$\frac{h \times g \times o}{e \times l \times 238}$$

$$\frac{p \times g \times o}{e \times l \times 103}$$

where o is the cost of oil, cents per gallon;
l the pounds per gallon of oil, and
g the pounds of oil per brake horsepower-hour.

79. With **Steam Drive** the cost of coal in cents per 1000 gallons of water is

$$\frac{h \times c \times 521}{d}$$

$$\frac{p \times c \times 1203}{d}$$

where d is the duty in foot-pounds per 1000 pounds of steam, evaporation 8 pounds water per pound coal,
c is the cost of 2000 lb. coal in cents.

80. Data for **Belt Drive**

Horsepower for single leather belts;

$$p = \frac{d\,w\,r}{3000}$$

Horsepower for double leather belts;

$$p = \frac{d\,w\,r}{1800}$$

where p is the horsepower;
d the diameter of the pulley in inches;
w the width of the belt in inches, and
r the revolutions per minute of the pulley.

The center-to-center distance between pulleys should be not less than 2½ times the diameter of the larger pulley. This distance may be decreased when an idler pulley is used.

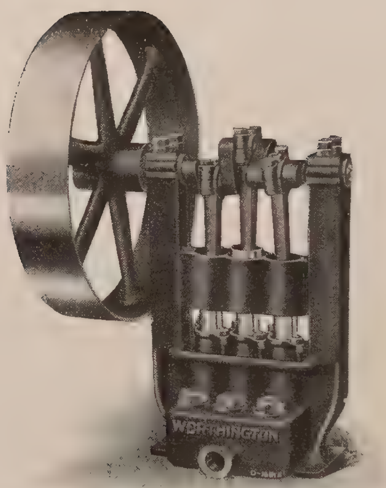


FIG. 26. Belt-driven pump with pulley mounted on crankshaft.

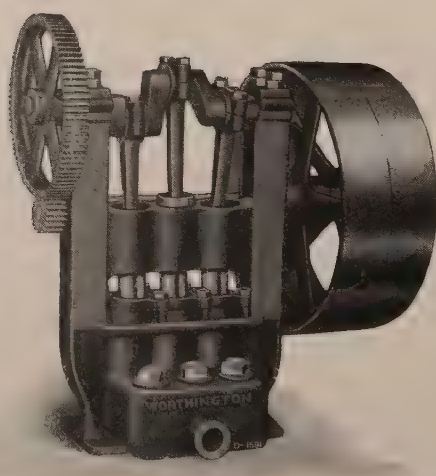


FIG. 27. Belt-driven pump showing tight and loose pulley with single gear reduction.

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

Single-acting plunger-pattern

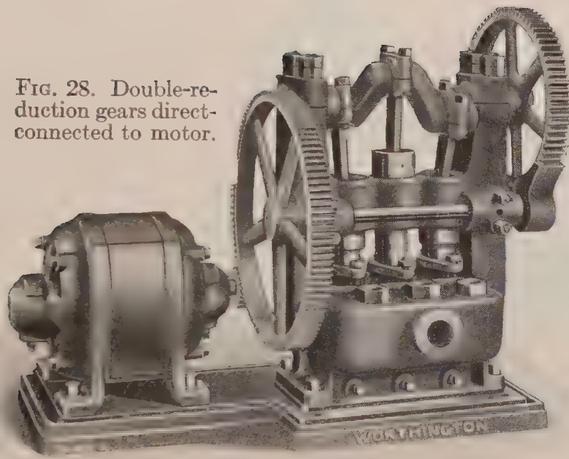


Fig. 28. Double-reduction gears direct-connected to motor.

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

Single-acting Plunger-pattern

Maximum working pressure: 200 lbs. per sq. in.

Figs. 26-27-28

85. TABLE OF SIZES AND DATA

Size, Inches		Displacement					Pipe Size, Inches		Tight Pulley, Inches					Approximate Dimensions in Ft. and In. with S. R. G. Pulley		
			Normal		Maximum				Fig. 27		Fig. 26					
Diameter of Plungers	Length of Stroke	Gal. per Rev.	Rev. per Min.	Gal. per Min.	Rev. per Min.	Gal. per Min.	Suction	Discharge	Diameter	Face	Diameter	Face	Ratio of Gearing	Length	Width	Height
1¼	2	0.032	80	2.55	86	2.75	1	¾	12	1¾	16	2½	5 : 1	1–1	1–1	1–8
1¾	2½	0.078	75	5.85	80	6.24	1	1	12	1¾	18	3½	5 : 1	1–4	1–3	1–10

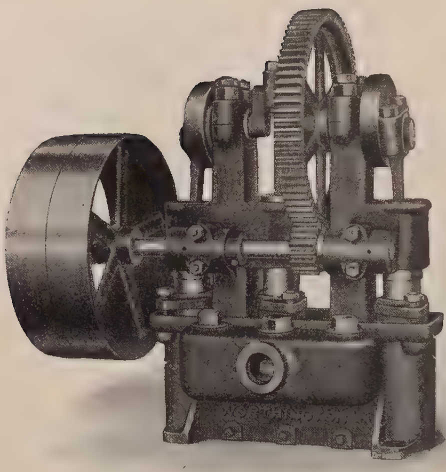


FIG. 29. Belt-driven pump with single gear reduction.

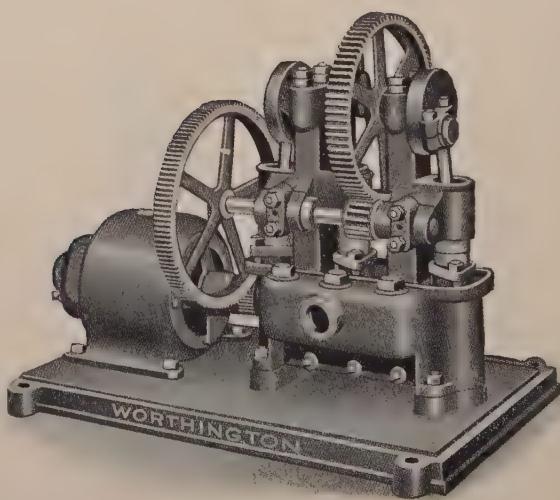


FIG. 30. Pump and motor mounted on a common base plate with double gear reduction.

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

Single-acting plunger-pattern

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

Single-acting Plunger-pattern

Maximum working pressure: 150 lb. per sq. in.

Figs. 29 and 30.

86. TABLE OF SIZES AND DATA

Size, Inches		Displacement					Pipe Sizes, Inches		Tight Pulley, Inches			Approximate Dimensions in Inches Belted Pump		
			Normal		Maximum									
Diameter of Plungers	Length of Stroke	Gal. per Rev.	Rev. per Min.	Gal. per Min.	Rev. per Min.	Gal. per Min.	Suction	Discharge	Diameter	Face	Ratio of Gearing	Length	Width	Height
2	3	0.122	50	6.1	75	9.15	1¼	1¼	12	2¼	5 : 1	18	15	23
3	4	0.367	50	18.3	68	24.95	2	1½	15	3½	5 : 1	24	19	30
4	4	0.652	50	32.6	68	44.33	2½	2	18	3½	5 : 1	34	24	37
4	6	0.979	50	49.0	58	56.78	3	2½	20	4½	5 : 1	35	26	45

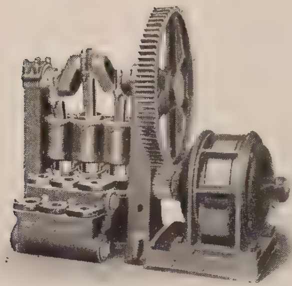


Fig. 31. Pump with single gear reduction and motor drive.

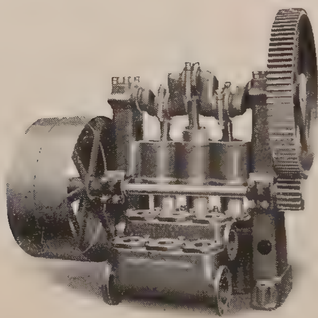
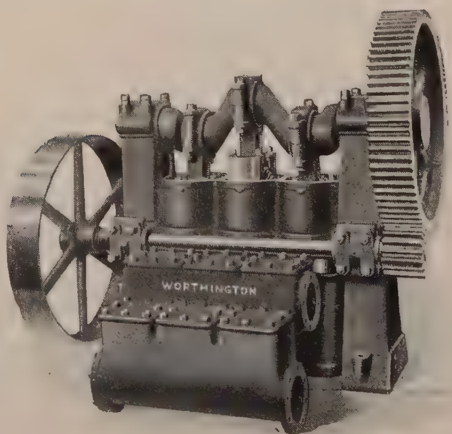


Fig. 32. Belt-driven pump with single gear reduction.

Fig. 33. Pump with single gear reduction and pulley for belt drive.



Worthington Vertical Triplex Power Pumps

Single-acting plunger-pattern

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

SINGLE-ACTING PLUNGER-PATTERN

Maximum working pressure: 200 lb. per sq. in.

FIGS. 31, 32 and 33.

87. TABLE OF SIZES AND DATA

Size, Inches		Displacement					Dia. Pipes, Inches		Tight Pulley, Inches		Ratio of Gearing	Approximate Dimensions in Ft. and In. S. R. G. T. Pulley		
			Normal		Maximum									
Diameter of Plungers	Length of Stroke	Gal. per Rev.	Rev. per Min.	Gal. per Min.	Rev. per Min.	Gal. per Min.	Suction	Discharge	Diameter	Face		Length	Width	Height
THESE SIZES SIMILAR TO FIGS. 31 AND 32.														
3	3	0.27	100	27.0	110	29.7	2	2	15	3½	5 : 1	2– 8	1– 9	2– 0
3½	4	0.50	80	40.0	88	44.0	3	2½	18	4½	5 : 1	3– 3	2– 0	3– 2
4	6	0.97	80	77.6	88	85.4	4	3	20	5½	5 : 1	3– 4	2– 4	3– 9
5	6	1.53	80	122.0	88	134.7	4	4	20	6½	5 : 1	3– 7	2– 9	4– 1
THESE SIZES SIMILAR TO FIG. 33.														
6	8	2.93	68	199.0	75	220.0	†5	†5	36	8½	5 : 1	4–10	3– 9	5– 7
8	8	5.22	68	355.0	75	391.0	†8	†6	42	7½	5 : 1	6– 0	4– 6	5– 8
9	10	8.26	68	495.0	75	545.0	†8	†8	54	9½	5 : 1	6–10	5– 3	6–10

8 and 10-in. stroke pumps have flanged openings.

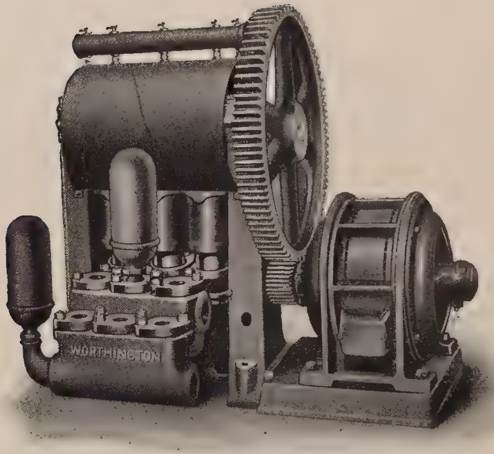
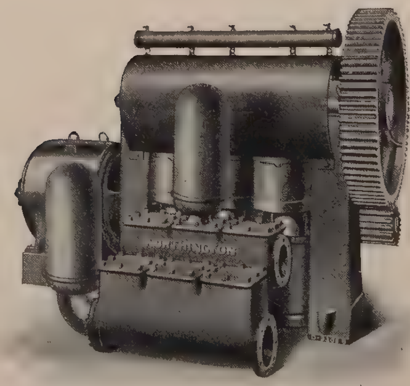


Fig. 34. Pump with single gear reduction and high-speed motor drive. Pumps having 6-in. stroke and under are of this style.

Fig. 35. Pumps of 8-in. stroke and over use this type of drive.



Worthington Vertical Triplex Power Pumps

Single-acting plunger-pattern with splash guards

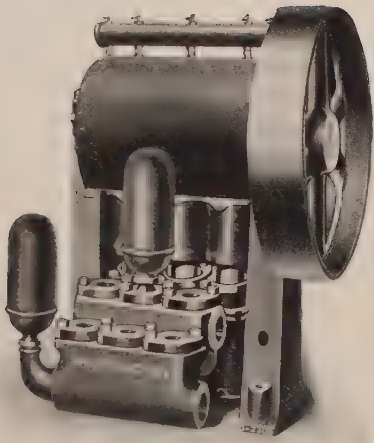


FIG. 36. Belt-driven pump showing the design for sizes having 6-in. stroke and less.

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

SINGLE-ACTING PLUNGER-PATTERN WITH SPLASH GUARDS

Maximum working pressure: 200 lb. per sq. in.

FIGS. 34, 35 and 36.

88. TABLE OF SIZES AND DATA

Size, Inches		Maximum Pressure	Displacement			Pipe Sizes, Inches		Tight Pulley, Inches		Approximate Dimensions Ft. and In.		
Diameter of Plungers	Length of Stroke		Gal. per Rev.	Rev. per Min.	Gal. per Min.	Suction	Discharge	Diameter	Face	Length	Width	Height
3	3	200	0.27	130	35	2	2	20	5½	2– 7	2– 1	2– 7
3½	4	200	0.50	120	60	4	3	30	6½	3– 7	2–10	3– 8
4	6	200	0.97	104	101	4	3	36	8½	4– 4	3– 2	4– 4
5	6	200	1.53	104	159	*5	*5	48	8½	4– 5	4– 0	5– 0
6	8	200	2.93	94	275	*8	*6	......	......	5– 8	3– 9	5– 7
8	8	200	5.22	94	490	*8	*8	......	......	6– 0	4– 6	5– 8
9	10	200	8.26	86	710	*10	*8	......	......	6–10	5– 3	6–10

* Flanged openings. All others tapped; 8- and 10-in. stroke pumps are furnished with single-reduction gear and flexible coupling drive.

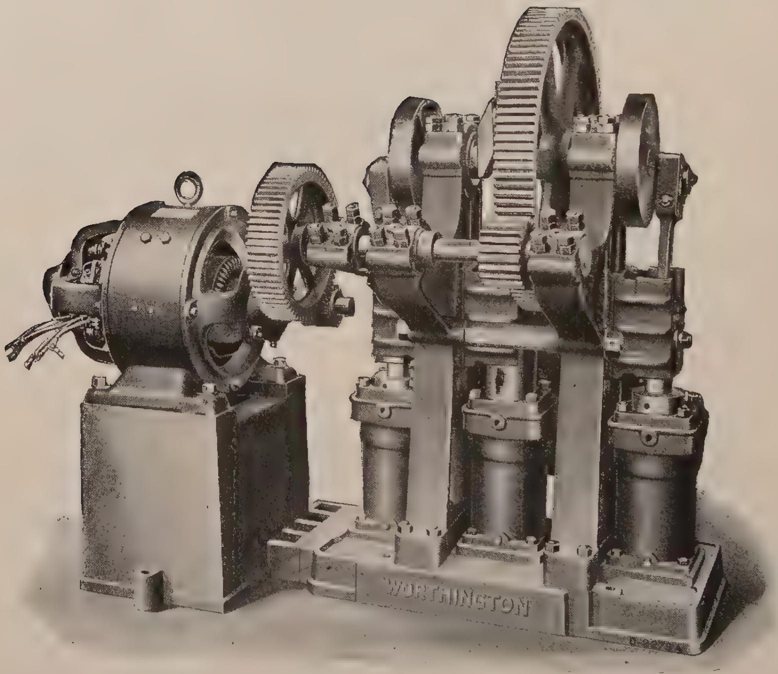


FIG. 37. Motor-driven type with double gear reduction.

WORTHINGTON VERTICAL TRIPLEX POWER PUMP

Single-acting plunger-pattern

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

Single-acting Plunger-pattern

Maximum working pressure: see table below

Pumps can be either belt-driven or direct-connected to motor.

Fig. 37.

89. TABLE OF SIZES AND DATA

Size, Inches			Displacement					Pipe Size, Inches		Pulleys, Inches			Approx. Dimensions in Ft. and In. S. R. G. T. Pulley		
				Normal		Maximum									
Diameter	Length of Stroke	Maximum Working Pressure	Gal. per Rev.	Rev. per Min.	Gal. per Min.	Rev. per Min.	Gal. per Min.	Suction	Discharge	Diameter	Face	Ratio of Gearing	Length	Width	Height
6	8	100	2.93	45	132	50	146	5	4	30	5½	4.91–1	3– 5	4–10	5–2
7	8	100	4.00	45	180	50	200	5	4	36	6½	4.66–1	3– 6	4– 4	5–6
8	8	125	5.22	45	235	50	261	6	5	36	8½	4.88–1	5– 1	5– 9	6–5
5	6	150	1.53	50	76	58	89	4	3	20	6½	4.91–1	3– 0	3–11	4–9
5	8	150	2.04	45	91	50	102	5	4	30	5½	4.91–1	3– 5	4– 5	4–9
6	8	150	2.93	45	132	50	146	5	4	36	6½	4.91–1	3–10	4– 2	5–2
7	8	150	4.00	45	180	50	200	5	4	36	8½	4.88–1	5– 1	5– 9	6–5
5	8	200	2.04	45	91	50	102	5	4	36	6½	4.91–1	3–10	4– 2	5–2
6	8	200	2.93	45	132	50	146	5	4	36	8½	4.88–1	5– 1	5– 6	6–5
4	6	250	0.98	50	49	58	57	4	3	20	6½	4.91–1	3– 0	3–11	4–9
4	8	250	1.30	45	58	50	65	4	3	36	6½	4.91–1	3–10	4– 2	5–2
5	8	250	2.04	45	91	50	102	5	4	36	8½	4.88–1	5– 1	5– 6	6–5

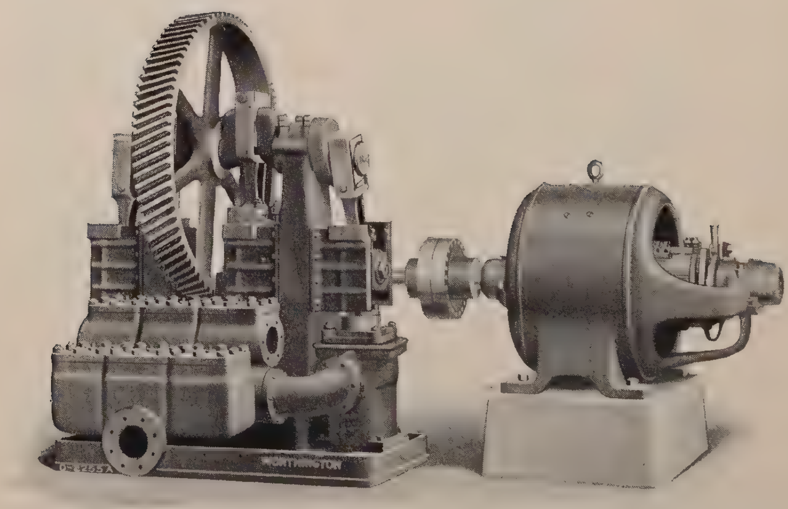


FIG. 38. Worthington Vertical Triplex Pump driven through flexible coupling by motor mounted on separate foundation.

WORTHINGTON VERTICAL TRIPLEX POWER PUMP

Single-acting plunger-pattern

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

Single-acting Plunger-pattern

Fig. 38.

Maximum working pressure: see table below.

Pumps can be either belt-driven or direct-connected to motor

90. TABLE OF SIZES AND DATA

Size, Inches		Maximum Pressure	Displacement			Pipe Sizes, Inches		Tight Pulley, Inches		Approximate Dimensions Pump with S. R. G. Pulley Ft. and In.		
Diameter of Plungers	Length of Stroke		Gal. per Rev.	Rev. per Min.	Gal. per Min.	Suction	Discharge	Diameter	Face	Length	Width	Height
9	10	100	8.26	45	372	8	6	36	8½	5– 3	6– 4	6– 5
10	10	100	10.20	45	459	10	8	42	7½	5– 7	6–11	6– 3
12	12	100	17.60	40	704	10	8	60	10½	7– 3	8– 1	8–10
14	12	100	24.00	40	959	12	12	66	13	9– 6	9– 4	9– 2
8	10	125	6.52	45	293	6	5	36	8½	5– 3	6– 4	6– 5
9	10	125	8.26	45	372	8	6	42	7½	5– 7	6– 5	6– 3
10	10	125	10.20	45	459	10	8	48	8½	5–11	6–11	6–11
11	12	125	14.80	40	592	10	8	60	10½	7– 3	8– 1	8–10
16	12	125	31.30	40	1252	12	12	84	15	11– 1	10–11	10– 1
7	10	150	4.99	45	224	6	5	36	8½	5– 3	5–11	6– 5
8	10	150	6.52	45	293	8	6	42	7½	5– 7	6– 0	6– 3
9	10	150	8.26	45	372	8	6	48	8½	5–11	6– 5	6– 5
10	10	150	10.20	45	459	10	8	54	9½	6–11	7–10	7– 5
12	12	150	17.60	40	704	10	8	66	13	9– 6	8– 8	9– 2
15	12	150	27.50	40	1102	12	12	84	15	11– 1	10–11	10– 1
11	12	175	14.80	40	592	10	8	66	13	9– 6	8– 8	9– 2
14	12	175	24.00	40	959	12	12	84	15	11– 1	10– 4	10– 1
7	10	200	4.99	45	224	8	6	42	7½	5– 7	5–11	6– 3
8	10	200	6.52	45	293	8	6	48	8½	5–11	6– 4	6– 5
9	10	200	8.26	45	372	8	6	54	9½	6–11	7– 3	7– 5
10	12	200	12.20	40	489	10	8	66	13	9– 6	8– 4	9– 2
7	10	250	4.99	45	224	8	6	48	8½	5–11	6– 4	6– 5
8	10	250	6.52	45	293	8	6	54	9½	6–11	7– 3	7– 5
9	12	250	9.91	40	396	8	6	66	13	9– 6	8– 4	9– 2
10	12	250	12.20	40	489	10	8	84	15	11– 1	9– 0	10– 1
11	12	250	14.80	40	592	10	8	84	15	11– 1	9– 0	10– 1
8	12	225	7.83	40	313	8	6	60	10½	7– 3	8– 4	8–10
12	12	225	17.60	40	704	10	8	84	15	11– 1	9– 0	10– 0

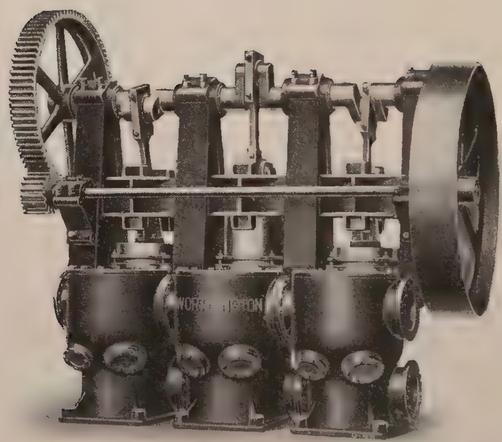


FIG. 39. Large, vertical single-acting Worthington Triplex Pumps, for moderate heads.

WORTHINGTON VERTICAL TRIPLEX POWER PUMP

Single-acting plunger-pattern

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

Single-acting Plunger-pattern

Maximum working pressure: see table below

Pumps can be either belt-driven or direct-connected to motor

Fig. 39.

91. TABLE OF SIZES AND DATA

Size, Inches		Maximum Pressure in Lb. per Sq. In.	Displacement			Pipe Sizes, Inches		Tight Pulley, Inches		Ratio of Gearing—Belted Pump	Approximate Dimensions Ft. and In. for Belted Pump		
Diameter of Plungers	Length of Stroke		Gal. per Rev.	Rev. per min.	Gal. per min.	Suction	Discharge	Diameter	Face		Length	Width	Height
8	8	100	5.22	75	392	8	6	42	6½	5 : 1	5– 2	4– 1	6– 6
9	12	100	9.90	60	594	10	8	58	9½	4.70 : 1	6– 4	5– 8	8– 9
10	12	100	12.30	60	738	10	8	58	9½	4.70 : 1	6– 4	5– 8	8– 9
11	12	100	14.80	60	889	12	10	58	9½	4.77 : 1	7– 0	6–11	9– 4
11	15	100	18.50	50	925	12	10	60	11	4.90 : 1	9– 0	7– 1	9– 6
15	15	100	34.00	50	1700	18	16	84	15	4.60 : 1	13–11	9– 1	11– 8
7	8	90	4.00	75	300	8	6	36	6½	5 : 1	5– 2	3– 6	5– 7
16	15	85	39.00	50	1950	18	16	84	15	4.60 : 1	13–11	9– 1	11– 8
9	8	80	6.61	75	496	8	6	42	6½	5 : 1	5– 2	4– 1	6– 6
12	12	80	17.60	60	1055	12	10	58	9½	4.77 : 1	7– 0	6–11	9– 4
12	15	80	22.00	50	1100	12	10	60	11	4.90 : 1	9– 0	7– 1	9– 6
17	15	75	44.20	50	2210	18	16	84	15	4.60 : 1	13–11	9– 1	11– 8
8	8	70	5.22	75	392	8	6	36	6½	5 : 1	5– 2	3– 6	5– 7
10	8	70	8.16	75	612	10	8	42	6½	5 : 1	5– 6	4– 3	6– 6
13	12	70	20.70	60	1242	12	10	58	9½	4.77 : 1	7– 3	6–11	9– 4
13	15	70	25.80	50	1290	12	10	66	11	4.90 : 1	9– 0	7– 4	9– 6
18	15	65	49.50	50	2475	18	16	84	15	4.60 : 1	13–11	9– 1	11– 8
14	12	60	24.00	60	1440	14	12	58	9½	4.77 : 1	7– 3	7– 6	9– 7
14	15	60	30.00	50	1500	14	12	66	11	4.90 : 1	9– 0	7–10	9– 8
19	15	60	55.20	50	2610	18	16	84	15	4.60 : 1	13–11	9– 1	11– 8
20	15	55	61.20	50	3060	18	16	84	15	4.60 : 1	13–11	9– 1	11– 8
15	12	50	27.60	60	1656	14	12	60	11	4.77 : 1	8– 4	7– 8	9– 7
15	15	50	34.00	50	1700	14	12	66	11	4.90 : 1	9– 0	7–10	9– 8
12	8	45	11.75	75	881	12	10	42	6½	5 : 1	6– 4	5– 4	6–10
16	12	45	31.30	60	1878	14	12	60	11	4.77 : 1	8– 4	7– 8	9– 7
16	15	45	39.00	50	1950	14	12	66	12	4.90 : 1	9– 8	7–10	9– 8
14	8	35	16.00	75	1200	12	10	42	6½	5: 1	6– 4	5– 4	6–10

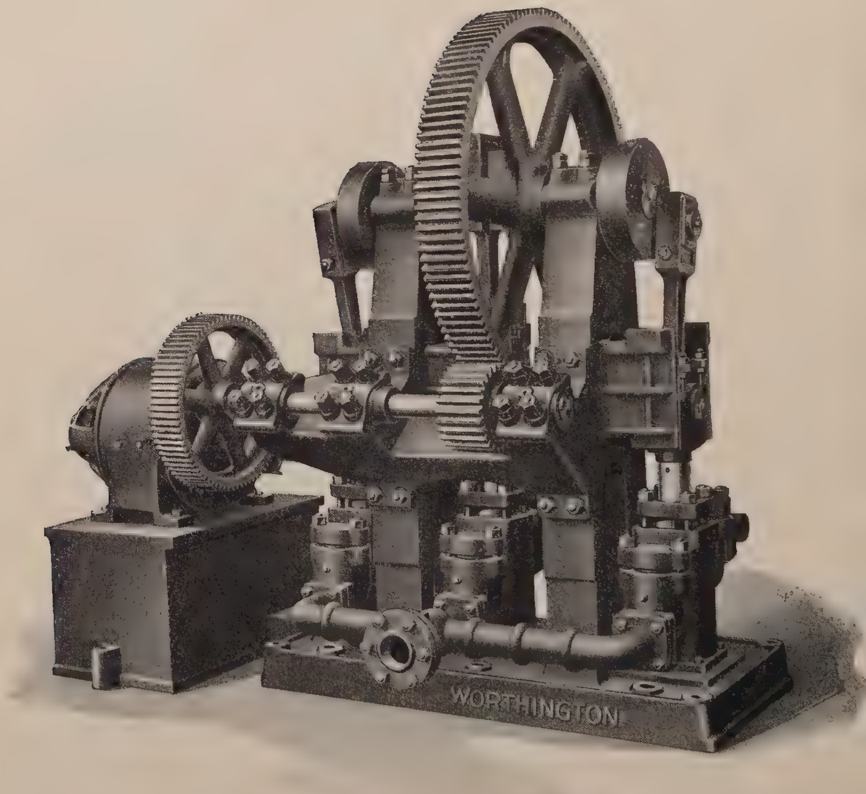


Fig. 40. Worthington Vertical Triplex Pumps, with double reduction gear. Driven by electric motor.

Worthington Vertical Triplex Power Pump

Single-acting plunger-pattern

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

Single-acting Plunger-pattern

Maximum working pressure: see table below.

Pumps can be either belt-driven or direct-connected to motor.

Fig. 40.

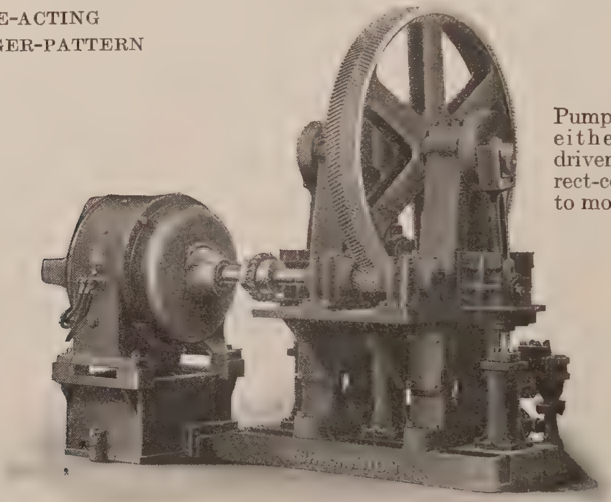
92. TABLE OF SIZES AND DATA

Size, Inches			Displacement					*Pipe Sizes		Tight Pulley, inches			Approx. Dimensions Pump with S. R. G. and Pulleys, Ft. In.		
				Normal		Maximum									
Diameter of Plunger	Lgth. of Stroke	Maximum Pressure Lb. per Sq. In.	Gal. per Rev.	Rev. per min.	Gal. per min.	Rev. per min.	Gal. per min.	Suction	Discharge	Diameter	Face	Ratio of Gearing	Length	Width	Height
1¾	6	1200	0.186	50	9.35	58	10.8	2	1½	20	6½	4.91 :	2– 8	3– 3	4–9
2	6	900	0.244	50	12.2	58	14.1	2	1½	20	6½	4.91 : 1	2– 8	3– 4	4–9
2½	6	600	0.382	50	19.1	58	22.1	2	1½	20	6½	4.91 : 1	2– 8	3– 8	4–9
3	6	400	0.550	50	27.5	58	31.9	2	1½	20	6½	4.91 : 1	2– 8	3– 8	4–9
2	8	950	0.326	45	14.6	50	16.3	2	1½	30	5½	4.91 : 1	3– 5	3–10	4–9
2	8	1250	0.326	45	14.6	50	16.3	2	1½	36	6½	4.91 : 1	3– 6	4– 2	4–9
2½	8	600	0.508	45	22.8	50	25.4	3	2	30	5½	4.91 : 1	3– 5	3–10	4–9
2½	8	800	0.508	45	22.8	50	25.4	3	2	36	6½	4.91 : 1	3– 6	4– 2	4–9
2½	8	1200	0.508	45	22.8	50	25.4	3	2	36	8½	4.88 : 1	5– 1	5– 2	6–5
3	8	400	0.734	45	33.0	50	36.7	3	2	30	5½	4.91 : 1	3– 5	4– 0	4–9
3	8	550	0.734	45	33.0	50	36.7	3	2	36	6½	4.91 : 1	3– 6	4– 2	4–9
3	8	850	0.734	45	33.0	50	36.7	3	2	36	8½	4.88 : 1	5– 1	5– 2	6–5
4	8	325	1.30	45	58.7	50	65.0	3	2	36	6½	4.91 : 1	3– 6	4– 2	4–9
4	8	475	1.30	45	58.7	50	65.0	3	2	36	8½	4.88 : 1	5– 1	5– 2	6–5
5	8	300	2.04	45	91.8	50	102.0	4	3	36	8½	4.88 : 1	5– 1	5– 2	6–5
3	10	1125	0.918	42	38.5	45	41.3	3	2	42	7½	5.07 : 1	5– 6	4– 3	6–3
4	10	625	1.63	42	68.0	45	73.4	4	3	42	7½	5.07 : 1	5– 6	4– 3	6–3
5	10	400	2.55	42	107.0	45	115.0	4	3	42	7½	5.07 : 1	5– 6	4– 3	6–3
5	10	500	2.55	42	107.0	45	115.0	4	3	48	8½	5.07 : 1	5–11	5–10	6–4
5	10	600	2.55	42	107.0	45	115.0	4	3	54	9½	4.78 : 1	6– 9	5– 5	7–5
6	10	275	3.67	42	154.0	45	165.0	5	4	42	7½	5.07 : 1	5– 6	4– 3	6–3
6	10	350	3.67	42	154.0	45	165.0	5	4	48	8½	5.07 : 1	5–11	5–10	6–4
6	10	425	3.67	42	154.0	45	165.0	5	4	54	9½	4.78 : 1	6– 9	5– 5	7–5

*Discharge pipe sizes are based on extra heavy pipe.

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

SINGLE-ACTING
PLUNGER-PATTERN



Pumps can be either belt-driven or direct-connected to motor.

FIG. 41. Pump with motor drive and single gear reduction.

93. TABLE OF SIZES AND DATA

Size, Inches			Displacement			Pipe Sizes, Inches		Tight Pulley, Inches			Approximate Dimensions in Ft. and In.		
Diameter of Plungers	Length of Stroke	Maximum Pressure in Lb. per Sq. In	Gal. per rev.	Rev. per min.	Gal. per min.	Suction	Discharge	Diameter	Face	Ratio of Gearing	Length	Width	Height
7	10	300	4.99	45	224	5	4	54	9½	4.77 : 1	7–3	8– 4	8–1
4½	12	1050	2.47	40	98	4	3½	66	13	5.00 : 1	9–6	8– 0	9–3
5	12	850	3.06	40	122	5	4	66	13	5.00 : 1	9–6	8– 0	9–3
6	12	600	4.40	40	176	5	4	66	13	5.00 : 1	9–6	9– 4	9–3
6	12	900	4.40	40	176	5	4	84	15	5.14 : 1	11–6	9– 7	10–2
7	12	450	6.00	40	240	6	5	66	13	5.00 : 1	9–6	9– 4	9–3
7	12	700	6.00	40	240	6	5	84	15	5.14 : 1	11–6	9– 7	10–2
8	12	325	7.83	40	313	6	5	66	13	5.00 : 1	9–6	10– 0	9–3
8	12	500	7.83	40	313	6	5	84	15	5.14 : 1	11–6	9– 9	10–2
9	12	400	9.91	40	396	7	6	84	15	5 14 : 1	11–6	9– 9	10–2
10	12	325	12.20	40	489	8	7	84	15	5.14 : 1	11–6	11–10	10–2

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

Single-acting Plunger-pattern
Special-service Forged Steel Liquid End for Heavy Pressure Work

94. Where **hydraulic presses** are used for baling or similar operations, the initial load is light and calls for a considerable volume of water at low pressure. As the operation progresses, the pressure increases and the volume of water required decreases, the final squeeze being obtained by a small volume at high pressure.

95. This service requires a special pump (Fig. 44) having one large plunger and two small plungers, operating together until a predetermined pressure is reached, at which point an automatic valve trips the suction valve supplying the larger plunger, temporarily cutting it out of service. The remaining small plungers continue in service and supply water at reduced capacity but at the higher pressure required, until the operation is completed.

96. Quotations will be promptly submitted on receipt of specifications.

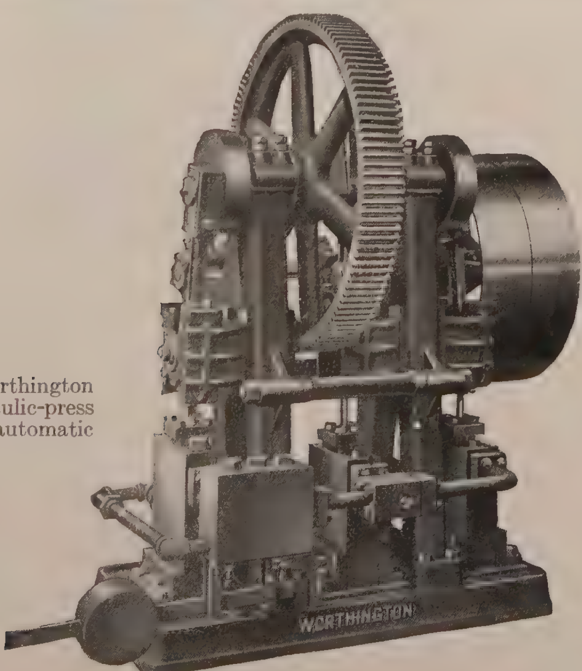


Fig. 42. Worthington Special Hydraulic-press Pump, with automatic cut-out valve.

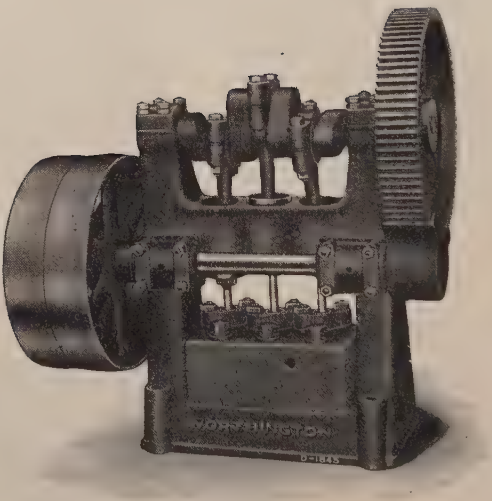


Fig. 43. The most popular drive for Worthington Medium Size Single-acting Triplex Pumps (tight and loose pulley with single-gear reduction).

Worthington Vertical Triplex Power Pump

Single-acting plunger-pattern

WORTHINGTON VERTICAL TRIPLEX POWER PUMP

SINGLE-ACTING PLUNGER-PATTERN

Maximum working pressure: see table below.

Pumps can be either belt-driven or direct-connected to motor.

FIG. 43.

97. TABLE OF SIZES AND DATA

Size, Inches		Maximum Pressure	Displacement					Pipe Sizes, Inches		Pulley Sizes, Inches				Std. Main Gear Ratio	Approximate Dimensions in Ft. and In. with S. R. G. and Pulley		
				Normal		Maximum				Fig. 42							
Diameter of Plungers	Length of Stroke		Gal. per Rev.	Rev. per Min.	Gal. per Min.	Rev. per Min.	Gal. per Min.	Suction	Discharge	Diameter	Face	Diameter	Face		Length	Width	Height
½	4	9800	0.010	95	.95	102	1.04	¾	¾	18	3½	36	6½	5 : 1	2–5	1–10	3–3
¾	4	4350	0.022	95	2.10	102	2.33	¾	¾	18	3½	36	6½	5 : 1	2–5	1–10	3–3
1	4	2450	0.040	95	3.86	102	4.15	¾	¾	18	3½	36	6½	5 : 1	2–5	1–10	3–3
1¼	4	1550	0.063	95	6.05	102	6.50	1	1	18	3½	36	6½	5 : 1	2–5	1–10	3–3
1½	4	1100	0.091	95	8.65	102	9.35	1¼	1¼	18	3½	36	6½	5 : 1	2–5	1–10	3–3
1¾	4	800	0.125	95	11.80	102	12.70	1¼	1¼	18	3½	36	6½	5 : 1	2–5	1–10	3–3
¾	6	8900	0.034	80	2.73	87	2.97	¾	¾	20	6½	36	6½	5 : 1	3–7	2– 5	4–1
1	6	5000	0.061	80	4.92	87	5.35	1	1	20	6½	36	6½	5 : 1	3–7	2– 5	4–1
1¼	6	3200	0.095	80	7.64	87	8.33	1	1	20	6½	36	6½	5 : 1	3–7	2– 5	4–1
1½	6	2225	0.137	80	11.00	87	11.90	1½	1¼	20	6½	36	6½	5 : 1	3–7	2– 5	4–1
1¾	6	1650	0.187	80	15.00	87	16.30	1½	1¼	20	6½	36	6½	5 : 1	3–7	2– 5	4–1
2	6	1250	0.244	80	19.50	87	21.20	1½	1¼	20	6½	36	6½	5 : 1	3–7	2– 5	4–1
1	8	7200	0.081	68	5.54	75	6.11	1¼	1	36	8½	...	....	5 : 1	5–0	3– 8	5–7
1¼	8	4650	0.127	68	8.65	75	9.54	1¼	1	36	8½	...	....	5 : 1	5–0	3– 8	5–7
1½	8	3200	0.183	68	12.40	75	13.70	1¼	1¼	36	8½	...	....	5 : 1	5–0	3– 5	5–7
2	8	1800	0.327	68	22.20	75	24.50	1½	1½	36	8½	...	....	5 : 1	5–0	3– 8	5–7

Maximum gear ratio with single reduction of gears and motor base drive is 10 to 1. For larger gear ratios use double reduction. With either of these drives, motors over 25 hp. should be furnished with outboard bearing.

Rawhide pinion cannot be furnished for single reduction of spur gear and motor base drive for the maximum conditions.

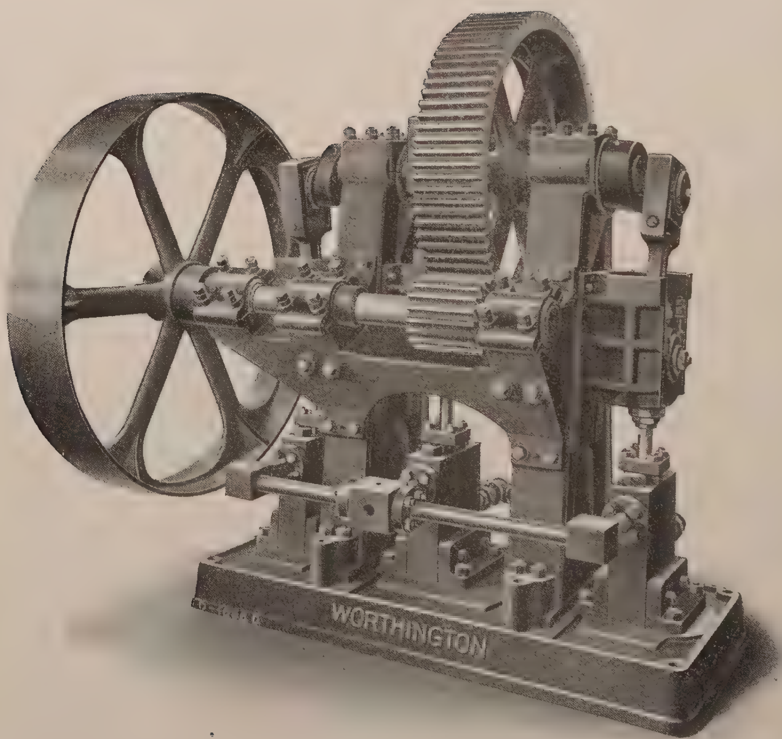


Fig. 44. Triplex single-acting pressure pump with single gear reduction.

Worthington Vertical Triplex Power Pump

Single-acting plunger-pattern

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

SINGLE-ACTING PLUNGER-PATTERN

Maximum working pressure: see table below.

Pumps can be either belt-driven or direct-connected to motor.

FIG. 44.

98. TABLE OF SIZES AND DATA

Size, Inches		Maximum Pressure in Lb. per Sq. In.	Displacement			Dia. Pipe, Inches		Tight Pulley, Inches		Ratio of Gearing	Approx. Dimensions in Ft. and In. S. R. G. and Pulley		
Diameter of Plungers	Length of Stroke		Gal. per Rev.	Rev. per Min.	Gal. per Min.	Suction	*Discharge	Diameter	Face		Length	Width	Height
1½	10	4500	0.229	68	15.6	2	1½	42	7½	5.07 : 1	5– 7	4–10	6–3
1½	10	5650	0.229	68	15.6	2	1½	48	8½	5.07 : 1	6– 0	4–11	6–5
1½	10	6800	0.229	68	15.6	2	1½	54	9½	4.77 : 1	7– 0	5–10	7–6
2	10	2550	0.408	68	27.8	2	1½	42	7½	5.07 : 1	5– 7	5– 1	6–3
2	10	3200	0.408	68	27.8	2	1½	48	8½	5.07 : 1	6– 0	5– 0	6–5
2	10	3800	0.408	68	27.8	2	1½	54	9½	4.77 : 1	7– 0	5–10	7–6
2½	10	1600	0.637	68	43.4	2½	2	42	7½	5.07 : 1	5– 7	5– 1	6–3
2½	10	2050	0.637	68	43.4	2½	2	48	8½	5.07 : 1	6– 0	5– 0	6–5
2½	10	2450	0.637	68	43.4	2½	2	54	9½	4.77 : 1	7– 0	5–10	7–6
3	10	1700	0.918	68	62.4	3	2	54	9½	4.77 : 1	7– 0	5–10	7–6
1½	12	9650	0.275	60	16.5	2	1½	60	15	5.00 : 1	8–10	6–10	9–3
2	12	5400	0.489	60	29.3	2	1½	60	15	5.00 : 1	8–10	6–10	9–3
2	12	8500	0.489	60	29.3	2	1½	60	21	5.14 : 1	10– 6	7– 2	10–0
2	12	10,000	0.489	60	29.3	2	1½	60	29	4.96 : 1	12– 0	8– 2	11–4
2½	12	3500	0.765	60	46.0	2½	2	60	15	5.00 : 1	8–10	6–10	9–3
2½	12	5500	0.765	60	46.0	2½	2	60	21	5.14 : 1	10– 6	7– 2	10–0
2½	12	7400	0.765	60	46.0	2½	2	60	29	4.96 : 1	12– 0	8– 2	11–4
3	12	2400	1.10	60	66.0	3½	2½	60	15	5.00 : 1	8–10	6–10	9–3
3	12	3750	1.10	60	66.0	3½	2½	60	21	5.14 : 1	10– 6	7– 2	10–0
3	12	5100	1.10	60	66.0	3½	2½	60	29	4.96 : 1	12– 0	8– 2	11–4
3½	12	1750	1.50	60	90.0	3½	2½	60	15	5.00 : 1	8–10	6–10	9–3
3½	12	2750	1.50	60	90.0	3½	2½	60	21	5.14 : 1	10– 6	7– 2	10–0
3½	12	3750	1.50	60	90.0	3½	2½	60	29	4.96 : 1	12– 0	8– 2	11–4
4	12	2100	1.95	60	117.0	4	3	60	21	5.14 : 1	10– 6	7– 2	10–0
4	12	2850	1.95	60	117.0	4	3	60	29	4.96 : 1	12– 0	8– 2	11–4
4½	12	1675	2.47	60	148.0	5	3	60	21	5.14 : 1	10– 6	7– 2	10–0
4½	12	2250	2.47	60	148.0	5	3	60	29	4.96 : 1	12– 0	8– 2	11–4
5	12	1800	3.06	60	183.0	5	3½	60	29	4.96 : 1	12– 0	8– 2	11–4
5½	12	1500	3.70	60	222.0	6	4	60	29	4.96 : 1	12– 0	8– 2	11–4

12-in. stroke pumps are provided with outboard bearings on the pinion shaft when pulley driven. *Discharge pipe sizes are based on double extra heavy pipe.

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

Single-acting plunger-pattern

STUFF PUMP

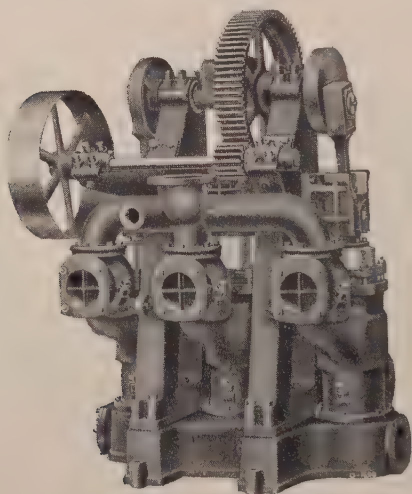


FIG. 45. Worthington Triplex Stuff Pump for paper-mill use.

99. TABLE OF SIZES AND DATA

Size, Inches		Maximum Pressure in Lb. per Sq. In.	Displacement				Pipe Sizes, Inches		Tight Pulley, Inches		Ratio of Gearing	Approximate Dimensions, Ft. and In.		
Diameter of Plungers	Length of Stroke		Gal. per Rev.	Rev. per Min.	Gal. per Min.	Tons of Dry Paper in 24 Hours	Suction	Discharge	Diameter	Face		Length	Width	Height
5	8	60	2.04	30	61	7 to 11	4	4	20	5½	5 : 1	5– 4	3– 2	6–
6	8	60	2.93	30	88	10 to 16	5	5	20	5½	5 : 1	5– 4	3– 2	6– 1
7	8	60	4.00	30	120	14 to 22	5	5	36	6½	5 : 1	5– 6	3–10	6– 6
														6
8	8	60	5.22	30	158	19 to 28	6	6	36	8½	5 : 1	5–10	3–10	6– 6
8	10	60	6.52	30	195	22 to 33	6	6	42	6½	5 : 1	5– 8	4– 4	7– 6
9	10	60	8.26	30	247	29 to 44	8	8	42	7½	5 : 1	5–10	4– 4	7– 8
														8
10	10	60	10.20	30	306	34 to 51	8	8	42	7½	5 : 1	5–10	4– 4	7– 8
10	12	60	12.24	27	330	40 to 60	8	8	48	8½	5 : 1	5–11	4– 7	7–10
12	12	60	17.62	27	476	57 to 85	10	10	54	11	5 : 1	7– 0	4–10	8– 0

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

Single-acting Plunger-pattern with Automatic Receiver

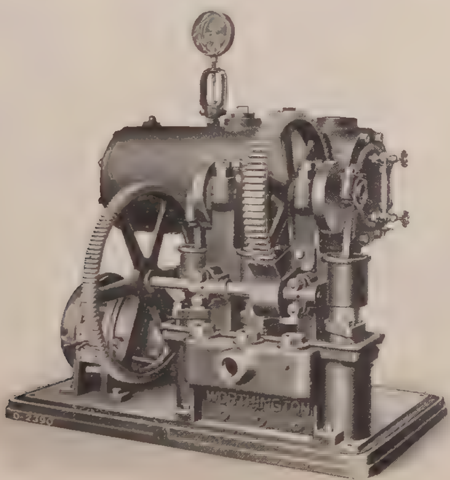


Fig. 46.

100. TABLE OF SIZES AND DATA

RECEIVERS				PUMPS			
Size Receiver Inches	Inlets No.	Inlets Dia. Inches	Recommended with Pumps Inches	Pump Size, Inches	Rev. per Min.	U. S. Gal. per Min.	Sq. Ft. of Radiating Surface Drained
13 x 30	3	2½	1¼ x 2 to 3 x 4	1¼ x 2	35	1.11	1,000
				1¾ x 2½	35	2.72	2,500
18 x 48	3	2½	3½ x 4 to 5 x 8	2 x 3	34	4.15	4,000
				3 x 3	34	9.35	8,000
				3 x 4	33	12.1	10,000
				3½ x 4	33	16.5	15,000
				4 x 4	33	21.5	20,000
				4 x 6	32	31.3	30,000
				5 x 6	32	49.0	45,000
				4 x 8	32	41.6	40,000
				5 x 8	33	67.3	65,000

These units are made for discharge pressures up to 250 lb. per sq. in. The design of pump varies slightly for different pressures, but the illustration is typical of the general arrangement.

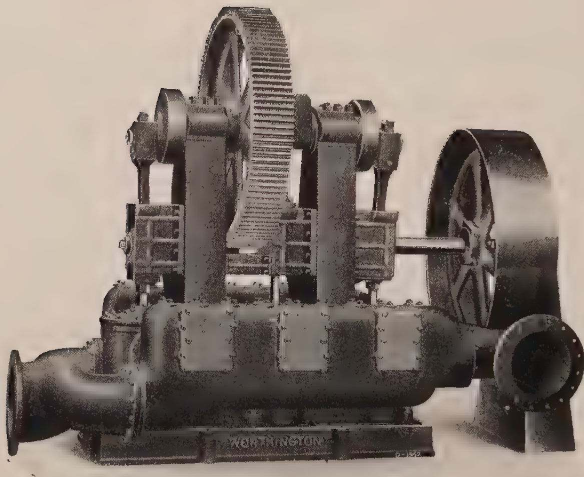


Fig. 47

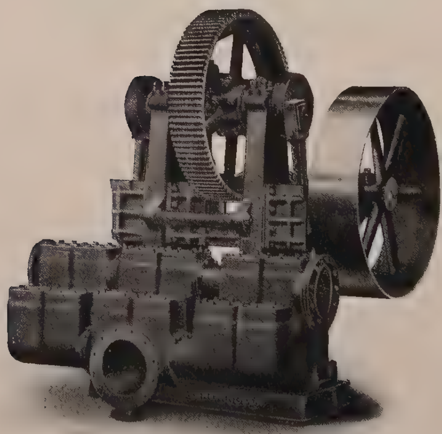


Fig. 48

Worthington Vertical Triplex Power Pumps

Double-acting piston-pattern

WORTHINGTON VERTICAL TRIPLEX POWER PUMPS

Double-acting Piston-pattern

Maximum working presssure: see table below

Figs. 47 and 48.

101. TABLE OF SIZES AND DATA*

Size, Inches		Maximum Pressure, Lb. per Square Inch	Displacement			Pipe Size, Inches		Tight Pulley, Inches		Ratio of Gearing Belted Pump	Approx. Dimensions Ft. and In., for Belted Pump		
Diameter of Pistons	Length of Stroke		Gal. per Rev.	Rev. per Min.	Gal. per Min.	Suction	Discharge	Diameter	Face		Length	Width	Height
6	8	75	5.74	45	258	6	5	36	6½	4.91:1	4– 5	5– 0	5– 7
6	8	175	5.74	45	258	6	5	42	7½	4.88:1	5– 8	5– 3	6– 5
7	8	125	7.87	45	353	7	6	42	7½	4.88:1	5– 8	5– 7	6– 5
8	8	100	10.31	45	463	8	7	42	7½	4.88:1	5– 0	5– 8	6– 5
7	10	75	9.55	42	401	8	7	42	6½	4.88:1	5– 0	5– 8	6– 5
8	10	60	12.74	42	535	8	7	42	6½	4.88:1	5– 0	5– 8	6– 5
7	10	150	9.55	42	401	8	7	48	12½	5.07:1	6– 4	6– 8	6–11
7	10	200	9.55	42	401	8	7	54	12½	5.07:1	6– 8	6– 9	7– 7
7	10	235	9.55	42	401	8	7	60	12½	4.78:1	8– 6	7– 6	8– 1
8	10	125	12.7	42	535	8	7	48	12½	5.07:1	6– 4	6– 8	6–11
8	10	150	12.7	42	535	8	7	54	12½	5.07:1	6– 8	6– 9	7– 7
8	10	175	12.7	42	535	8	7	60	12½	4.78:1	8– 6	7– 6	8– 1
9	10	50	16.2	42	680	10	8	42	6½	4.88:1	5– 5	7– 1	7– 0
9	10	100	16.2	42	680	10	8	48	12½	5.07:1	7– 2	6– 8	6–11
9	10	125	16.2	42	680	10	8	54	12½	5.07:1	7– 5	6– 9	7– 7
9	10	150	16.2	42	680	10	8	60	12½	4.78:1	8– 6	7– 6	8– 1
10	10	75	20.0	42	843	10	10	48	12½	5.07:1	7– 2	6– 8	6–11
10	10	100	20.0	42	843	10	10	54	12½	5.07:1	7– 5	6– 9	7– 7
10	10	125	20.0	42	843	10	10	60	12½	4.78:1	8– 6	7– 6	8– 1
9	12	200	19.3	40	773	10	10	84	16	5.00:1	9– 8	8–10	9–10
10	12	150	23.9	40	959	10	10	84	16	5.00:1	9– 8	8–10	9–10
10	12	225	23.9	40	959	10	10	84	24	5.14:1	12– 0	9– 1	11– 2
11	12	75	29.0	40	1160	14	12	66	12½	5.29:1	9– 2	8– 0	9– 4
11	12	135	29.0	40	1160	14	12	84	16	5.00:1	9– 8	9– 2	9–10
11	12	200	29.0	40	1160	14	12	84	24	5.14:1	12– 0	9– 5	11– 2
12	12	75	34.6	40	1385	14	12	66	12½	5.29:1	9– 2	8– 0	9– 4
12	12	100	34.6	40	1385	14	12	84	16	5.00:1	9– 8	9– 2	9–10

Pumps having pulleys 60 in. or over are provided with outboard bearings.

*Concluded on following page.

101. TABLES OF SIZES AND DATA—Concluded

Size, Inches		Maximum Pressure, Lb. per Square Inch	Displacement			Pipe Size, Inches		Tight Pulley, Inches		Ratio of Gearing Belted Pump	Approx. Dimensions Ft. and In., for Belted Pump		
Diameter of Pistons	Length of Stroke		Gal. per Rev.	Rev. per Min.	Gal. per Min.	Suction	Discharge	Diameter	Face		Length	Width	Height
12	12	175	34.6	40	1385	14	12	84	24	5.14:1	12– 0	9– 5	11– 2
14	12	50	47 3	40	1893	16	14	66	12½	5.29:1	0– 5	8– 6	9– 7
14	12	75	47.3	40	1893	16	14	84	16	5.00:1	11– 0	9– 8	10–10
14	12	125	47.3	40	1893	16	14	84	24	5.14:1	13– 0	9–11	11– 2
15	12	50	54.3	40	2172	18	16	66	12½	5.29:1	13– 6	7–11	9– 7
*15	12	75	54.3	40	2172	18	16	84	16	5.00:1	13– 6	9– 1	10–10
*15	12	100	54.3	40	2172	18	16	84	24	5.14:1	14– 6	9– 4	11– 2
16	12	40	61.8	40	2475	18	16	66	12⅓	5.29:1	13– 6	7–11	9– 7
*16	12	60	61.8	40	2475	18	16	84	16	5.00:1	13– 6	9– 1	10–10
*16	12	100	61.8	40	2475	18	16	84	24	5.14:1	14– 6	9– 4	11– 2

*These sizes have valve chests on both sides, as in Fig. 47.

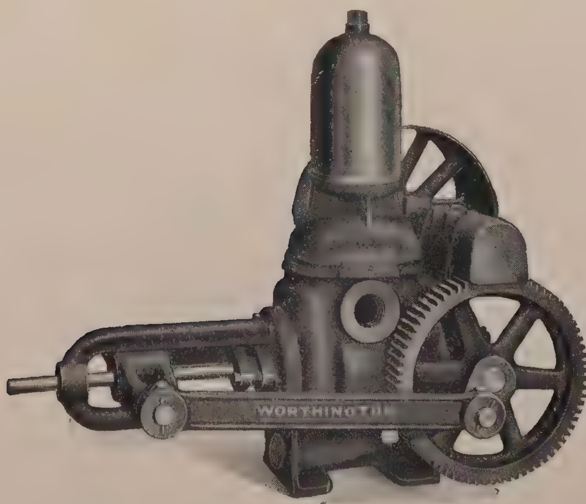


Fig. 49. Model "A" Worthington Horizontal Single Double-acting Power Pump for belt drive with single gear reduction.

WORTHINGTON HORIZONTAL SINGLE POWER PUMPS

Packed-Piston Pattern

103. Model "A" pumps, as shown by Fig. 49, are adapted for service against pressures up to 75 lb. per sq. in. The simple and compact design makes this type primarily suited for general water supply service for farms, country estates, clubs, hotels, dairies and in similar positions where a pump of comparatively low first cost is desired.

104. Model "A" pumps are recommended for use in connection with the pressure tank system of water distribution.

105. Gear and pinion are made of steeline, the teeth being machine cut; cylinder is brass-lined and piston rod is brass-covered on the fluid end. A feature which appeals most forcibly to those who have used the types of pump previously obtainable for rural service is the simplicity which permits the examination of the entire valve service, by the removal of a single nut located at a point most easy of access; namely, on top of the air chamber. These pumps are better suited for intermittent than for continuous service, and cannot be used for handling hot water.

106. TABLE OF SIZES AND DATA

Fig. 49.

Size		Maximum Pressure in Lb. per Sq. In.	Displacement			Diam. Pipes		Tight Pulley			Ratio of Gearing	Approx. Dimensions Ft. and In.	
Diameter of Piston	Length of Stroke		Gal. per Rev.	Rev. per Min.	Gal. per Min.	Suction	Discharge	Diameter	Face	*Belt		Length	Width
3	5	75	0.30	50	15	1½	1½	12	3	S	5:1	3–1	1–8
4	5	75	0.54	50	27	2½	2½	16	4	S	5:1	3–8	2–5
5	5	75	0.85	50	42	2½	2½	16	4	S	5:1	3–8	2–5
6	6	50	1.47	50	73	3	3	20	4	S	5:1	3–8	2–5
6	12	75	2.94	40	117	4	4	24	4½	S	5:1	6–0	2–9

*S indicates single belt.

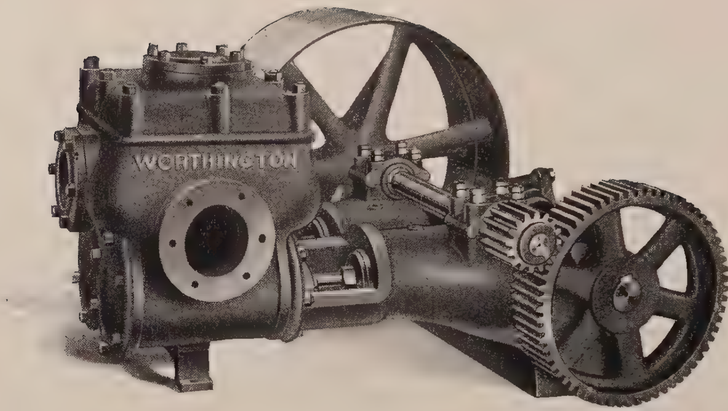


FIG. 50.

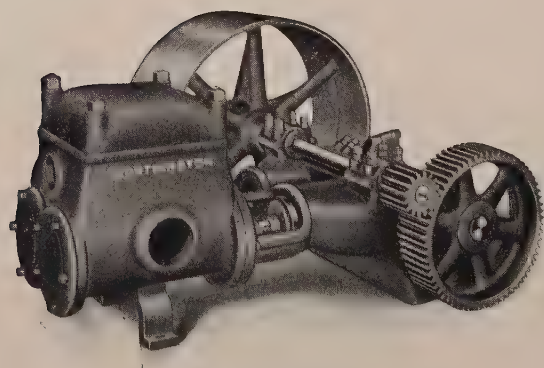


FIG. 51.

WORTHINGTON HORIZONTAL DUPLEX DOUBLE-ACTING POWER PUMPS

Packed-piston pattern

WORTHINGTON HORIZONTAL DUPLEX DOUBLE-ACTING POWER PUMPS

Packed-piston Pattern

Figs. 50 and 51.

106. TABLE OF SIZES AND DATA

Size		Maximum Water Pressure in Lb. per Sq. In.	Displacement			Diam. Pipes		Tight Pulleys			Ratio of Gearing	Approx. Dimensions Ft. and In.	
Diameter of Pistons	Length of Stroke		Gal. per Rev.	Rev. per Min.	Gal. per Min.	Suction	Discharge	Diameter	Face	*Belt		Length	Width
3	4	150	0.49	75	36	2	1½	15	3½	D	4.07:1	2– 9	1–10
3¾	4	120	0.76	75	57	2½	1½	18	3½	D	5:1	3– 4	2– 2
4	6	150	1.30	60	78	2½	1½	26	5½	D	5:1	3– 8	2– 6
5	6	100	2.04	60	122	3	2	26	5½	D	5:1	3–11	2– 9
6	6	75	2.93	60	175	4	3	26	5½	D	5:1	4– 3	2– 9
7	6	100	4.00	60	240	6	5	36	6½	D	4.76:1	4– 9	2–11
8	6	85	5.22	60	313	6	5	36	6½	D	4.76:1	4– 9	2–11
8½	6												

* D indicates double belt.

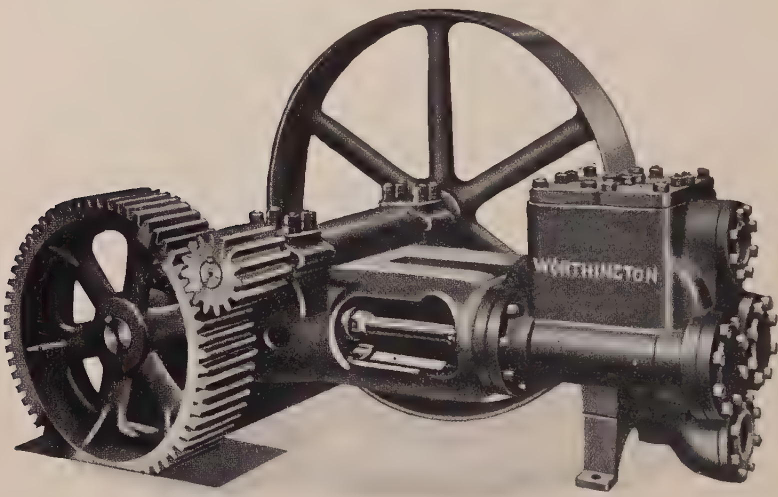


Fig. 52. Belt-driven pump with single gear reduction.

WORTHINGTON HORIZONTAL DUPLEX DOUBLE-ACTING POWER PUMPS

Piston Pattern for High Pressures

107. TABLE OF SIZES AND DATA

Size		Maximum Pressure in Lb. per Sq. In.	Capacity			Diameter Pipes		Ratio of Gearing	Approximate Dimensions Ft. and In.	
Diameter of Plungers	Length of Stroke		Gal. per Rev.	Rev. per Min.	Gal. per Min.	Suction	Discharge		Length	Width
3½	12	400	1.86	50	93	3½	3	5.78:1	6– 6	3– 9
4¼	12	400	2.81	50	140	4	3	4.06:1	6– 8	4– 4
5	12	300	3.83	50	191	4	3	4.06:1	6– 8	4– 4

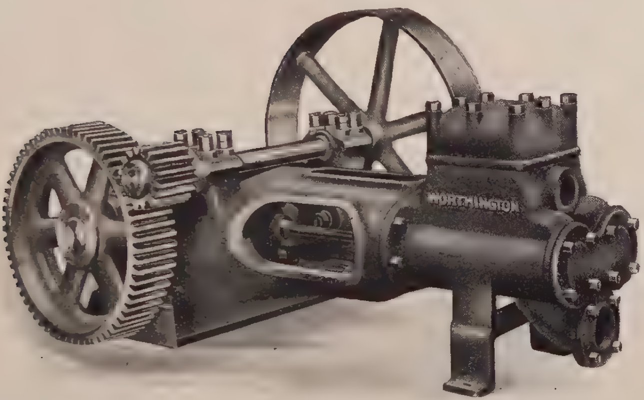


FIG. 53

WORTHINGTON HORIZONTAL DUPLEX DOUBLE-ACTING POWER PUMPS

PACKED-PISTON PATTERN

108. TABLE OF SIZES AND DATA

Size			Displacement			Diam. Pipes		Tight Pulley				Approx. Dimensions Ft. and In.	
Diameter of Pistons	Length of Stroke	Maximum Water Pressure in Lb. per Sq. In.	Gal. per Rev.	Rev. per Min.	Gal. per Min.	Suction	Discharge	Diameter	Face	*Belt	Ratio of Gearing	Length	Width
4½	12	188	3.31	50	165	4	3	36	6½	D	4.06:1	6–2	4–0
5	12	153	4.08	50	204	4	3	36	6½	D	4.06:1	6–2	4–0
5	12	200	4.08	50	204	5	4	42	7½	D	4.06:1	6–3	4–3
6	12	200	5.87	50	294	5	4	42	7½	D	4.06:1	6–3	4–3
7	12	156	8.00	50	400	†6	5	42	7½	D	4.06:1	6–7	4–3
8	12	120	10.44	50	522	†6	5	42	7½	D	4.06:1	6–7	4–3
8½	12	126	11.80	50	590	†6	5	42	7½	D	4.06:1	6–7	4–3
9	12	95	13.21	50	660	8	7	42	7½	D	5:1	7–8	4–8
9½	12	85	14.72	50	736	8	7	42	7½	D	5:1	7–8	4–8
10	12	77	16.32	50	815	8	7	42	7½	D	5:1	7–8	4–8
10½	12	69	18.00	50	900	8	7	42	7½	D	5:1	7–8	4–8

* D indicates double belt.
† These sizes have side suction.

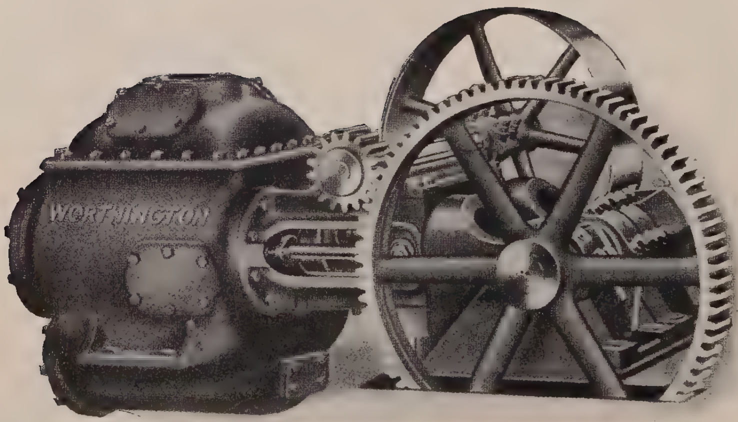


Fig. 54.

Worthington Horizontal Duplex Double-acting Power Pumps

Packed-piston pattern

WORTHINGTON HORIZONTAL DUPLEX DOUBLE-ACTING POWER PUMPS

Packed-Piston Pattern

Fig. 54.

109. TABLE OF SIZES AND DATA

Size			Displacement			Diam. Pipes		Tight Pulley				Approx. Dimensions Ft. and In.	
Diameter of Pistons	Length of Stroke	Maximum Water Pressure in Lbs. per Sq. In.	Gal. per Rev.	Rev. per Min.	Gal. per Min.	†Suction	Discharge	Diameter	Face	*Belt	Ratio of Gearing	Length	Width
6	12	150	5.87	60	352	7	6	48	8½	D	5:1	8–3	4–6
7	12	150	8.00	60	480	8	7	48	8½	D	5:1	8–3	4–6
8	12	119	10.45	60	626	10	8	48	8½	D	5:1	8–6	4–6
9	12	94	13.21	60	793	10	8	48	8½	D	5:1	8–6	4–6
‡9	12	150	13.21	60	793	10	8	60	12½	D	5:1	12–9	7–9
10	12	76	16.32	60	980	12	10	48	8½	D	5:1	8–6	4–6
‡10	12	127	16.32	60	980	12	10	60	12½	D	5:1	12–9	7–9
‡11	12	105	19.74	60	1185	12	12	60	12½	D	5:1	12–6	7–9
11	12	136	19.74	60	1185	12	12	60	15	D	5:1	12–9	8–5
‡12	12	88	23.5	60	1410	14	12	60	12½	D	5:1	12–6	7–9
12	12	115	23.5	60	1410	14	12	60	15	D	5:1	12–9	8–5
‡14	12	65	32.0	60	1920	16	14	60	12½	D	5:1	13–6	8–0
14	12	84	32.0	60	1920	16	14	60	15	D	5:1	13–9	8–5

* D indicates double belt.

† 10-inch stroke pumps are fitted with side suction

‡ Yokes cast integral with frames on these pumps.

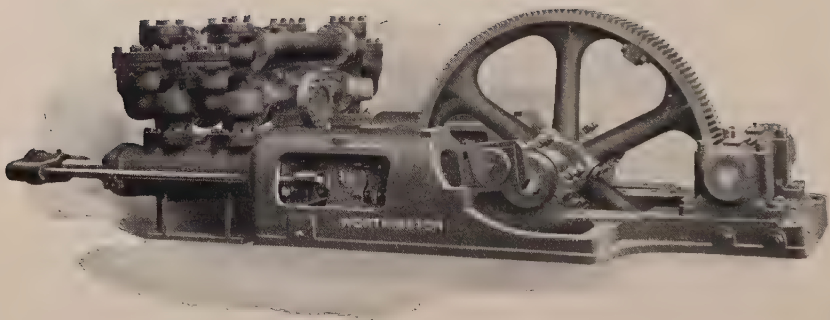


Fig. 55. Pump with single reduction gears for direct-connection to engine or motor.

Worthington Horizontal Duplex Double-acting Power Pumps

Outside-packed-plunger pot-valve pattern

WORTHINGTON HORIZONTAL DUPLEX DOUBLE-ACTING POWER PUMPS

OUTSIDE-PACKED-PLUNGER POT-VALVE PATTERN

FIG. 55.

110. TABLE OF SIZES AND DATA

Size		Maximum Water Pressure in Lb. per Sq. In.	Displacement			Diameter Pipes		Ratio of Gearing	Approx. Dimensions Ft. and In.	
Diameter of Plungers	Length of Stroke		Gal. per Rev.	Rev. per Min.	Gal. per Min.	Suction	†Discharge		Length	*Width
3	12	850	1.47	50	73	3½	3	4.06:1	9-8	4-6
3½	12	625	2.00	50	100	3½	3	4.06:1	9-8	4-6
3½	12	885	2.00	50	100	3½	3	4.71:1	10-4	4-8
4	12	675	2.60	50	130	5	4	4.71:1	10-4	4-8
4½	12	535	3.30	50	165	5	4	5.00:1	10-4	4-8
5	12	433	4.08	50	204	6	5	4.71:1	10-8	4-8
4	12	800	2.60	50	130	5	4	5:1	11-0	7-0
4½	12	630	3.30	50	165	5	4	5:1	11-0	7-0
4½	12	820	3.30	50	165	5	4	5:1	12-0	7-6
5	12	510	4.08	50	204	6	5	5:1	11-0	7-0
5	12	660	4.08	50	204	6	5	5:1	12-0	7-6
5½	12	420	4.93	50	246	6	5	5:1	11-0	7-0
5½	12	550	4.93	50	246	6	5	5:1	12-0	7-6
5½	12	756	4.93	50	246	6	5	5:1	14-9	7-2
6	12	354	5.88	50	294	6	5	5:1	12-2	6-6
6	12	460	5.88	50	294	6	5	5:1	12-8	6-6
6	12	640	5.88	50	294	6	5	5:1	14-9	7-2
6½	12	302	6.89	50	345	6	6	5:1	12-2	6-6
6½	12	393	6.89	50	345	6	6	5:1	12-8	6-6
6½	12	542	6.89	50	345	6	6	5:1	14-9	7-2
7	12	338	8.00	50	400	8	6	5:1	12-8	6-6
7	12	470	8.00	50	400	8	6	5:1	14-9	7-2
8	12	360	10.40	50	520	8	6	5:1	15-1	7-2

* Width given includes pinion shaft extension but does not include outboard bearing.

† Discharge pipe sizes are based on extra heavy pipe.

WORTHINGTON HORIZONTAL DUPLEX DOUBLE-ACTING POWER PUMPS

Outside-packed Plunger Forged Liquid-end Pattern

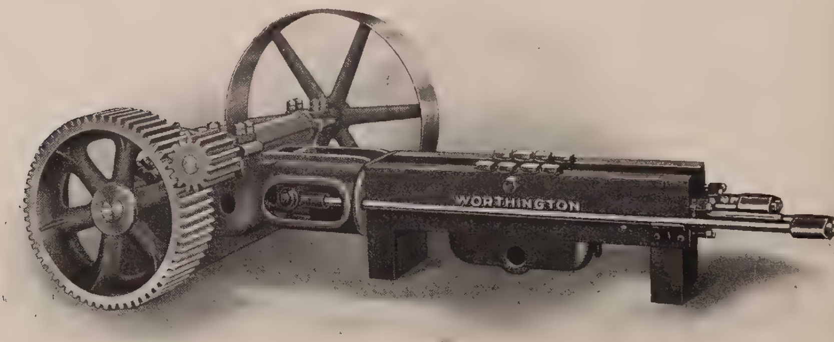


FIG. 56

111. TABLE OF SIZES AND DATA

Size		Maximum Water Pressure in Lb. per Sq. In.	Displacement			Pipe, Sizes		Tight Pulley		Ratio of Gearing of Belted Pump	Approximate Dimensions Ft. and In.	
Diameter of Plungers	Length of Stroke		Gal. per Rev.	Rev. per Min.	Gal. per Min.	Section	*Discharge	Diameter	Face		Length	Width
1	12	7640	0.163	60	9.78	1¼	1¼	54	8½	4.06:1	10– 8	4– 6
1	12	10000	0.163	60	9.78	1¼	1¼	60	9½	4.71:1	11– 2	4– 9
1¼	12	4880	0.255	60	15.3	1¼	1¼	54	8½	4.06:1	10– 8	4– 6
1¼	12	6930	0.255	60	15.3	1¼	1¼	60	9¼	4.71:1	11– 2	4– 9
1½	12	3400	0.368	60	22.0	1½	1½	54	8½	4.06:1	10– 8	4– 6
1½	12	4810	0.368	60	22.0	1½	1½	60	9½	4.71:1	11– 2	4– 9
1¾	12	2490	0.500	60	30.0	2	1½	54	8½	4.06:1	10– 8	4– 6
1¾	12	3530	0.500	60	30.0	2	1½	60	9½	4.71:1	11– 2	4– 9
2	12	1910	0.652	60	39.1	2½	2	54	8½	4.06:1	10– 8	4– 6
2	12	2700	0.652	60	39.1	2½	2	60	9½	4.71:1	11– 2	4– 9
2¼	12	1510	0.824	60	49.4	2½	2	54	8½	4.06:1	10– 8	4– 6
2¼	12	2140	0.824	60	49.4	2½	2	60	9½	4.71:1	11– 2	4– 9
2½	12	1220	1.02	60	61.	3	2½	54	8½	4.06:1	10– 8	4– 6
2½	12	1730	1.02	60	61.	3	2½	60	9½	4.71:1	11– 2	4– 9
2¾	12	1430	1.23	60	74.	3	2½	60	9½	4.71:1	11– 2	4– 9

* Discharge pipe sizes are based on double extra heavy pipe.

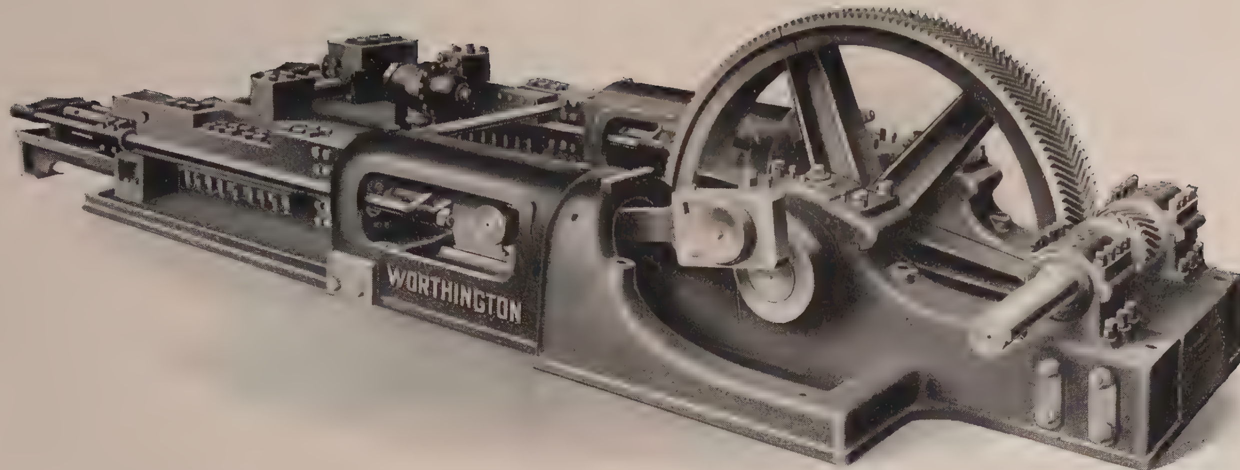


Fig. 57. Worthington Horizontal Duplex Power Pumps, with forged steel liquid-end for high pressure.

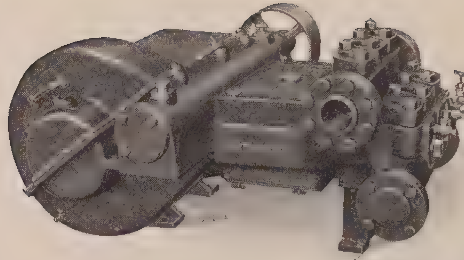


FIG. 58 Tight pulley for belt drive from engine, motor or other power source.

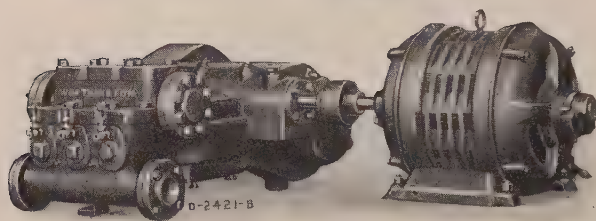


FIG. 59 Direct coupled to motor. This outfit may be mounted on skid or on permanent foundation .Can also be furnished with cast-iron or channel-iron base under pump and motor

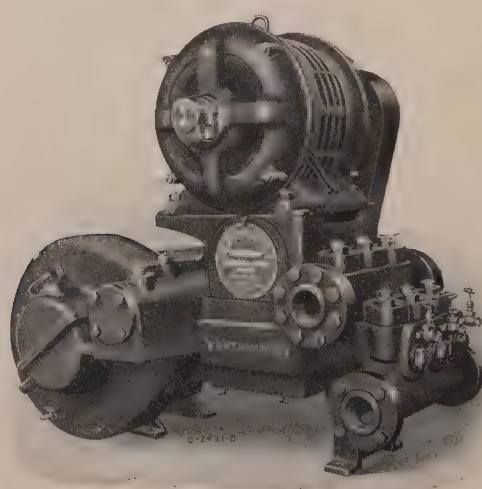


FIG. 60 Chain drive. Motor mounted over the pump frame.

WORTHINGTON HORIZONTAL TRIPLEX SINGLE-ACTING ENCLOSED-CRANK-CASE POWER PUMPS.

Self-oiling, dirt-proof, semi-portable.
Outside-packed-plungers, totally enclosed.

WORTHINGTON HORIZONTAL TRIPLEX POWER PUMPS

Single-Acting Enclosed Crankcase

Maximum working pressure: see table below

Figs. 58, 59 and 60

112. These pumps are adaptable for general service for capacities from 38 to 108 U. S. gals. per min., and pressures up to 500 lb. per sq. in. under an operating speed of 200 r.p.m. with a 6 to 1 gear ratio (this gear ratio is standard and not subject to change), which makes these pumps most desirable for direct connection to a 1200-r.p.m. motor or engine, without the use of belt or gear reduction. Slower speeds may be used with corresponding reduction in pump speed.

113. For special service the liquid end can be cast separate from the power end, and the parts coming in contact with the liquid can be of such non-corrosive metal as the requirements may demand. Likewise, other liquid-end parts, such as the plungers, the valve service, and all parts coming in contact with the liquid, can be furnished of materials best adapted for the contemplated service.

114. TABLE OF SIZES AND DATA

Size in Inches		Maximum Pressure	Maximum Displacement			Brake Horse Power Required to Pump Against Various Pressures (Pounds per Square Inch)					Shipping weight (lbs.)	Gear Ratio
dia. of plungers	length of stroke	lbs. per sq. inch	U.S. gals. per rev.	rev. per min.	U.S. gals. per min.	500	400	350	300	200		Main Gears Standard, Not subject to change
2¾	5	500	.387	200	77.5	28.8	23.7	21.2	18.4	12.5	1900	
.....	.....	.....	.....	175	67.7	25.6	21.0	18.9	16.3	11.2	.....	6
.....	.....	.....	.....	150	58.0	22.0	18.5	16.4	14.5	10.2	.....	
.....	.....	.....	.....	125	48.3	19.0	15.4	13.6	11.9	8.6	.....	
.....	.....	.....	.....	100	38.7	14.5	11.8	11.2	9.4	6.95	.....	to
3¼	5	350	.537	200	107.5	.....	.....	28.4	24.7	17.3	1900	
.....	.....	.....	.....	175	93.9	.....	.....	25.0	21.6	14.9	.....	
.....	.....	.....	.....	150	80.5	.....	.....	21.6	18.7	13.0	.....	1
.....	.....	.....	.....	125	67.1	.....	.....	18.3	16.0	11.2	.....	
.....	.....	.....	.....	100	53.7	.....	.....	14.9	13.0	9.3	.....	

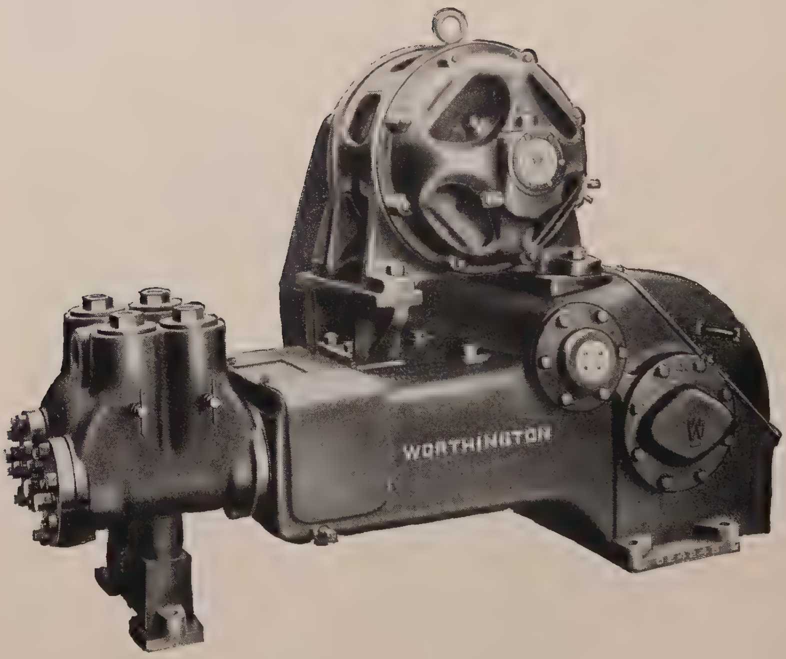


Fig. 61. Double-reduction gear drive with motor mounted over the pump frame. Also furnished with single-reduction gear for overhead chain or Texrope drive and for conventional belt drive.

Worthington Horizontal Duplex Double-acting Enclosed Crank-case Power Pump.

Self-Oiling, dirt-proof, semi-portable, piston-pattern, totally enclosed.

WORTHINGTON HORIZONTAL DUPLEX POWER PUMPS

Double-Acting Enclosed Crankcase

Fig. 61

115. Worthington horizontal, duplex, totally-enclosed, double-acting power pumps are especially adaptable for general service within the pressure and capacity range stated below. Their staunch construction, their dust- and sand- proof design, their positive self-oiling qualities, their semi-portability, and other desirable features render them particularly suitable and satisfactory for oil-fields and allied service. On road construction work and for similar purposes, they give most excellent results

117. TABLE OF SIZES AND DATA

Size in Inches		Max. Pres.	Displacement*					Horse-Power Required		
						bbls. (42 gal.)		The following motor sizes can be used up to the pressures stated (lbs. per sq. inch.)		
diam. of pistons	length of stroke	lbs. per square inch	U.S. gals. per rev.	rev. per min.	U.S. gals. per min.	per hour	per 24 hours	30 h.p.	40 h.p.	50 h.p.
2½	10	1400	.71	60	42.5	60	1455	900	1200	1400
3	10	1000	1.09	60	65.0	92	2225	600	800	1000
3½	10	750	1.51	60	91	130	3120	425	570	950
4	10	575	2.03	60	122	174	4180	320	425	575
4½	10	450	2.62	60	157	224	5380	250	330	450
5	10	350	3.25	60	195	278	6685	200	270	350
								(900 RPM Motors)		

*Displacements allow for piston rods.

SECTION III-A

Deep-Well Pumps

(Figures refer to paragraph numbers)

SECTION III-A

Deep-Well Pumps

1. Two interesting and useful types of pumps for deep-well service developed by Worthington are the Axiflo (Reg. U. S. Pat. Off.) and the Coniflo (Reg. U. S. Pat. Off.)

WORTHINGTON **AXIFLO** PUMP

(Reg. U. S. Pat. Off.)

2. Briefly, the Axiflo Pump consists of a vertical casing with a shaft through its center. On this shaft are mounted the required number of impellers. The action of these impellers is similar to that of a ship's propeller. The impellers, operating on a vertical shaft, pick up the water and force it vertically upward, inside of the casing and around the shaft. To eliminate eddy-currents and consequent waste of power, a set of discharge vanes is placed above each impeller. These check the whirling or rotary motion, and convert its kinetic energy into pressure, forcing the water upward in parallel instead of helical flow.

3. A typical impeller is shown in Fig. 1. This is a one-piece casting, finished smooth all over.

Fig. 1. Typical impeller for Worthington Axiflo Pump.

4. All dimensions are calculated for maximum efficiency, and the impeller is tested to check performance and flow constants. The impeller is locked rigidly in place on the pump shaft but the device used is such that it can readily be removed if necessary.

5. A direct-connected motor-driven unit is shown in Fig. 2.

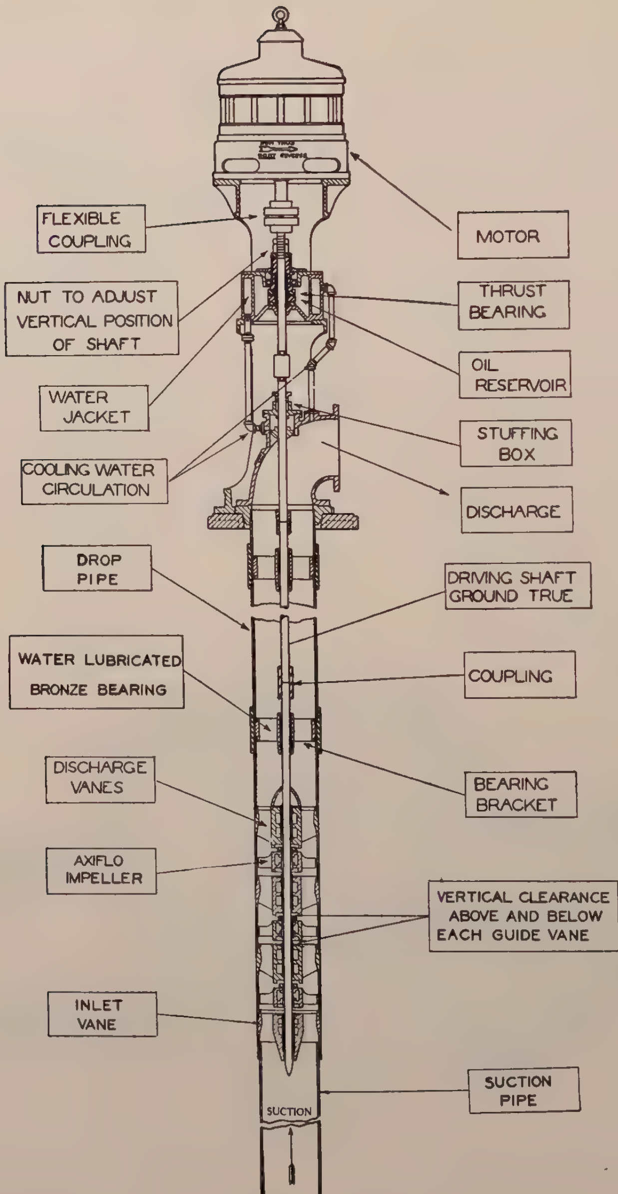


Fig. 2.

SECTIONAL VIEW OF WORTHINGTON AXIFLO PUMP.

6. As many impellers and sets of discharge vanes as may be required are used. This number depends on the depth of the well. Each impeller does its share of the work and passes the water along to the next. The operation of the pump is similar to that of a

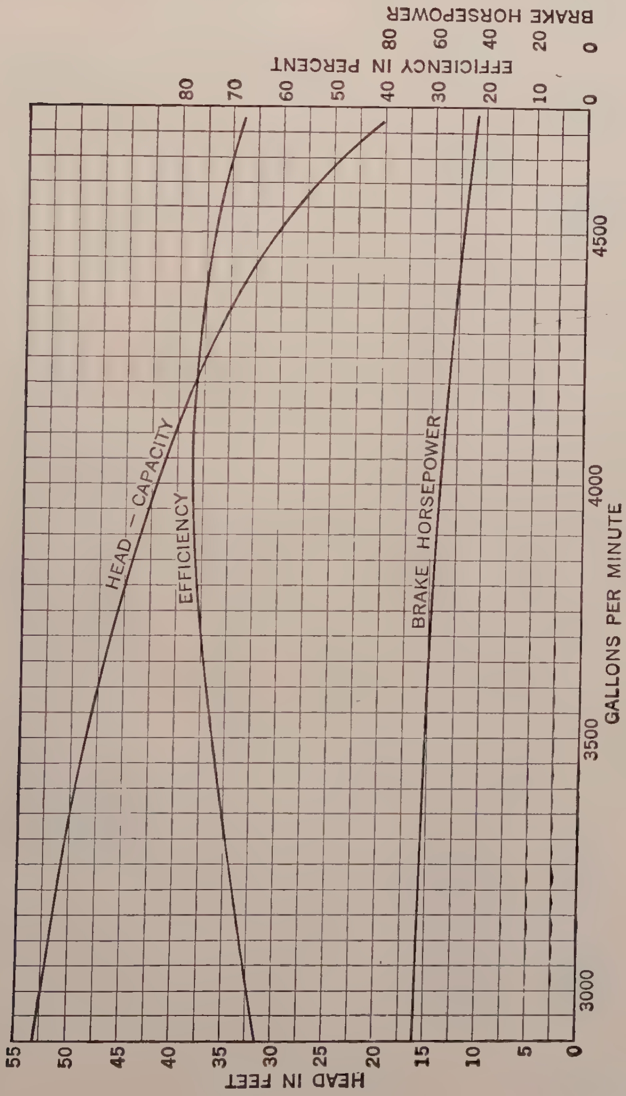


Fig. 3.

Characteristic Curves for No. 18 Worthington Axiflo Pump.

multi-stage centrifugal pump with the flow axial instead of radial. The performance curve of a typical AXIFLO pump as shown by Fig. 3 bears considerable resemblance to similar performance curves as worked out from tests on high-class centrifugal pumps.

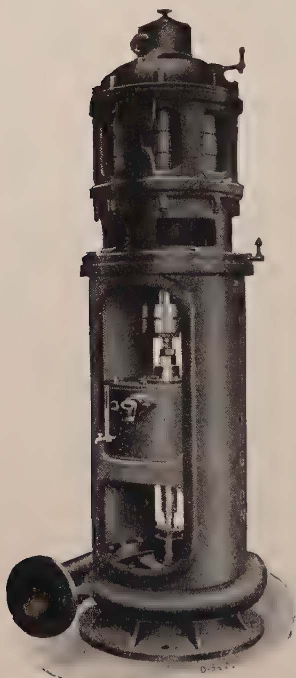


FIG. 4. Worthington AXIFLO driving head with self-contained Centrifugal Booster Pump.

7. As shown in the preceding illustrations, the pump is delivering the water at or near the surface of the ground. In cases where it is desired to discharge the water against a considerable head, a centrifugal booster pump is provided in the driving head (Fig. 4) and direct-connected to the pump shaft. This patented feature is a most valuable addition, enabling the pump to build up a high pressure for fire-protective purposes, or to supply a tank or reservoir at any reasonable distance away from or above the location of the driving head.

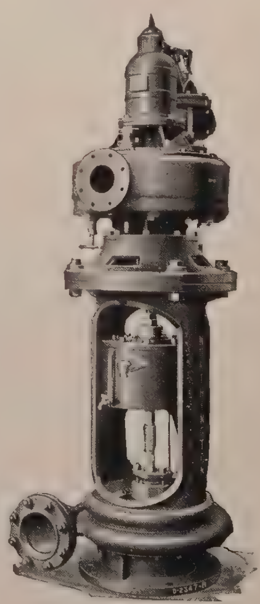


Fig. 5. Worthington AXIFLO driving head and Booster Pump with steam-turbine drive.

8. **Design.** — The Worthington AXIFLO pump has been designed accurately and carefully along scientific lines as a velocity machine. All the refinements common to high class velocity machines have been incorporated in the construction. It is not to be confused in construction or in the principle of operation with the ordinary screw pump.

9. The **driving head** is arranged to carry the motor, Fig. 6, steam turbine, Fig. 5, or belt pulley, Fig. 7, as the case may be, and to provide for proper access to the booster

element or the discharge elbow. There is a coupling between the discharge elbow or booster unit and the thrust bearing, removable when the pump is, for any reason, to be dismantled. This coupling has only to be taken off and the entire power head including the motor can be lifted off as a unit without dismantling. In this respect, the AXIFLO pump is superior to any other of the deep-well pumps, as they all require more or less complete dismantling of the head in order to get at the liquid end.

10. The whole driving-head mechanism is built around the thrust bearing, which must carry not only the water load on the impellers in the well, but the dead weight of the shaft, shaft couplings and impellers. At first glance it would appear that the logical solution of the thrust-bearing problem would be a ball thrust. Experiments extending over some two years at the Deane Works have fully

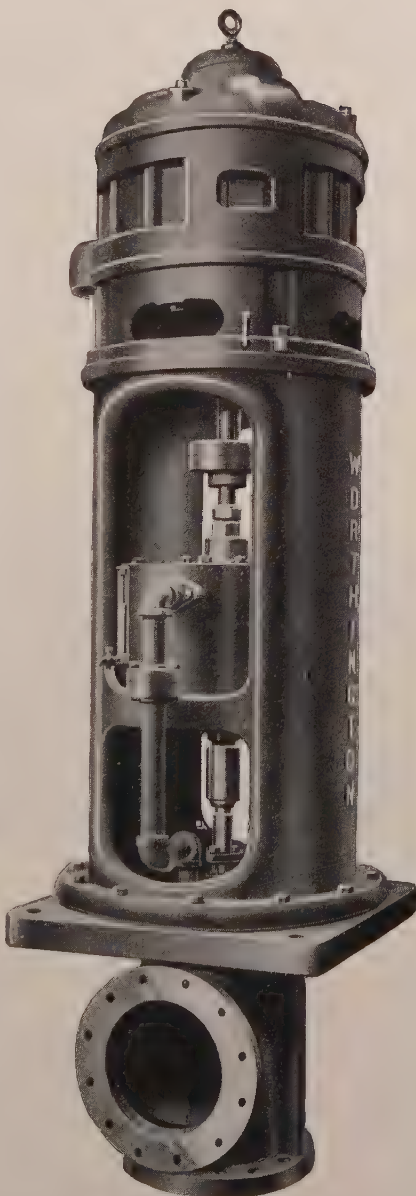


FIG. 6. Worthington AXIFLO driving head equipped with vertical motor.

proved that none of the commercial ball thrusts is trustworthy under the very heavy loads and high speeds at which the AXIFLO pumps are required to operate.

11. A **thrust bearing** (Fig. 8 and Fig. 9) similar in many respects to the Kingsbury which has been so successful in handling the enormous loads and high speeds on the thrusts of torpedo-boat destroyers and on similar marine service has been adopted for the AXIFLO pump. The stator segments are cast in one ring instead of separately. The principle of using a stator and rotor ring with provision for positively introducing lubricant in copious volume between the faces by means of radial grooves, and causing the lubricant to pass up over tapered approaches, is retained.

FIG. 7. Worthington AXIFLO Pump equipped for belt drive, under test.

12. This bearing is actually a **fluid step,** the entire load being carried on a film of oil. The bearing is submerged in a reservoir with passages so cut that there is a self-induced active circulation of lubricant at all times when the pump is running. The oil reservoir is of sufficient size so that under ordinary operating conditions it is not necessary to renew the oil supply oftener than once in six months or even a year.

13. To further insure satisfactory operation the oil reservoir on all of these pumps from the largest down to the smallest is water jacketed, as is shown in the diagram. A novel method of supply-

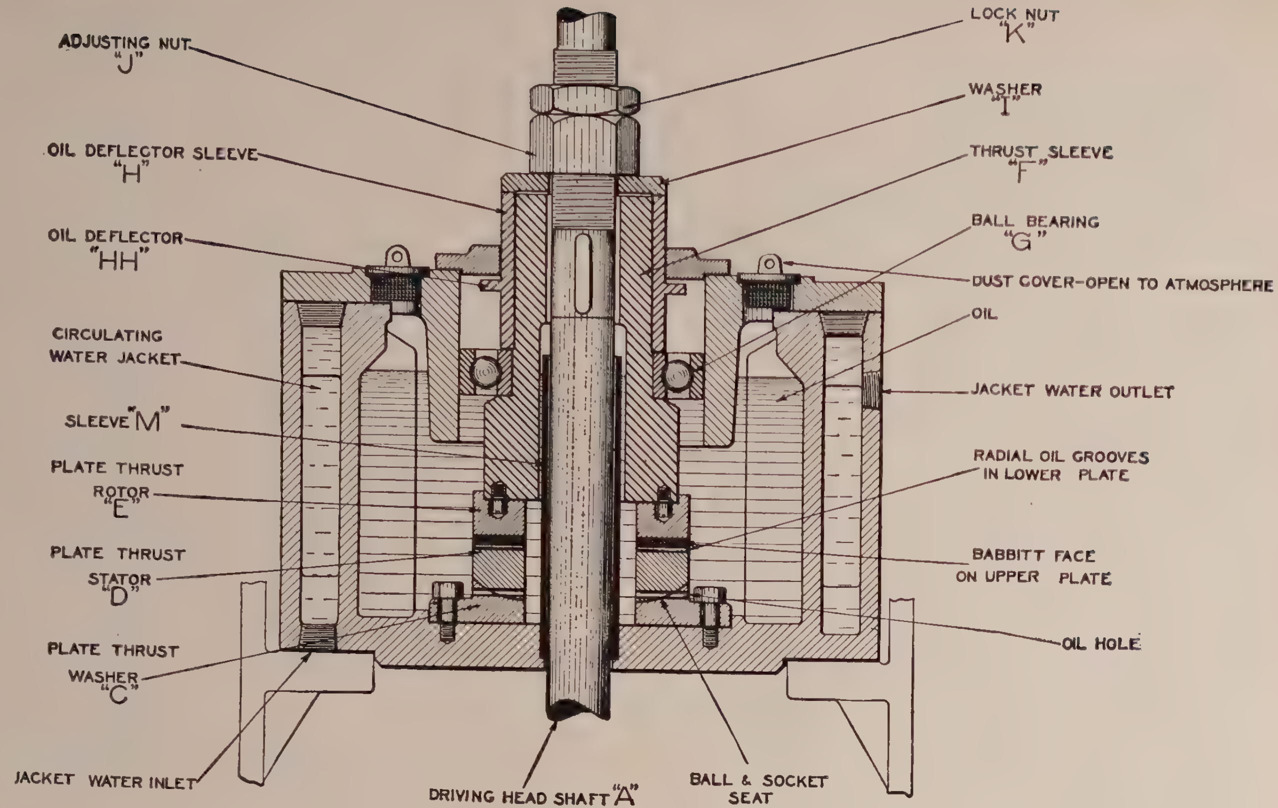


Fig. 8.
Thrust Bearing Used in Worthington Axiflo Pump for Motor Drive.

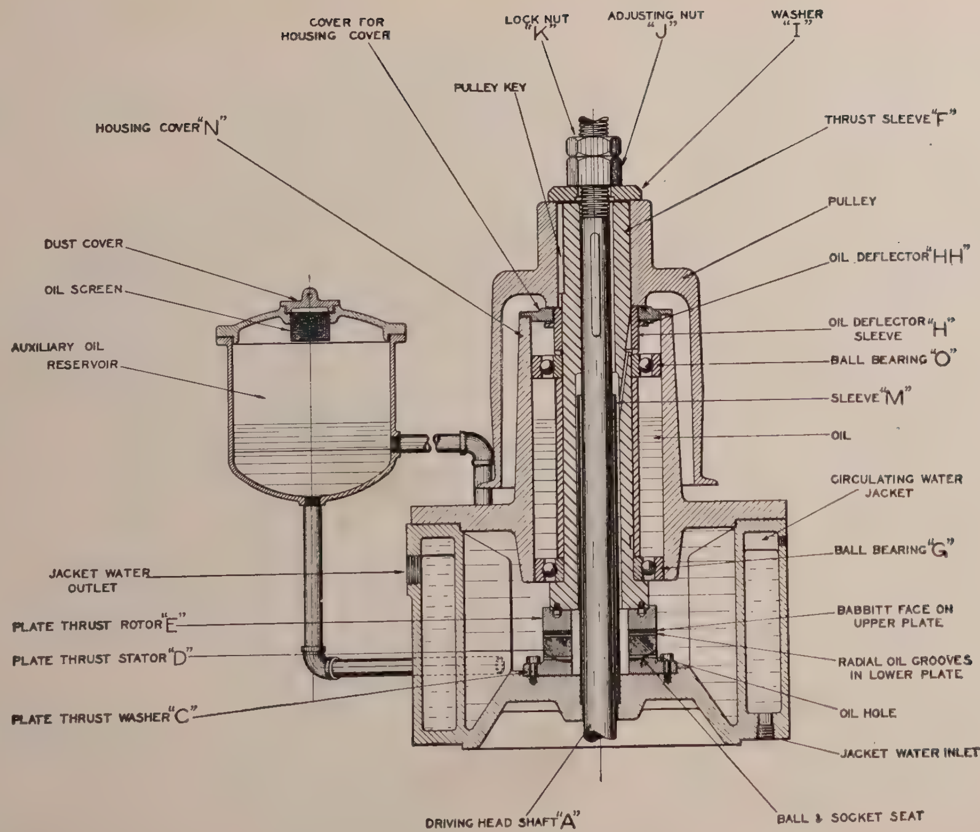


Fig. 9.

Thrust Bearing Used in Worthington Axiflo Pump for Belt Drive.

ing water to this jacket is employed, a small Axiflo impeller being located in the discharge elbow, elevating the water into the jackets, whence it passes up, around, out and back into the discharge elbow, thereby insuring continuance of circulation without any loss of water. In the type of pump with the booster, the jacket-water problem is simpler, an adequate supply being taken from the discharge and returned to the suction side of the centrifugal impeller.

14. It is, of course, in the **liquid end** that the great claim of superiority of the Axiflo Pump lies. The flow of water through the impellers is, in general, parallel with the shaft and we have therefore called the pump an Axiflo (derived from "axial flow").

15. The Axiflo type gives maximum capacities from wells of given sizes which are practically double the maxima for deep-well centrifugal pumps.

16. The pitch of the **impeller** blade (See Fig. 1) increases from the outer to the inner edge in proper proportion so as to give a uniform velocity to the passing water from the tip to the hub, thus avoiding differential velocities with resulting variations in pressure and cross flow which can only result in loss of efficiency. The blade is not a helix, the construction being so modified as to give shockless entrance and gradual acceleration to the water of the impeller.

17. The Axiflo impellers are of course more expensive to make than are the screw-type impellers, in that the pattern cannot be twisted out of a mold, owing to their changing pitch, but the increase of efficiency more than compensates for the increased cost.

18. The water is taken from the blades into a set of curved **guide or discharge vanes,** so designed as to convert the velocity into pressure head and bring the flow into vertical line for the next impeller with a minimum of impact and friction losses.

19. The **shaft** of the Worthington Axiflo Pump is of high-grade steel turned and ground. The shaft **bearings** in the Axiflo pumps are steady bearings with removable bushings of special leaded bronze. The shaft is in tension and it is only necessary for the bearings to prevent vibration.

20. The distance between the bearings is worked out so that the shaft will not operate at or near any of its critical speeds under any condition of service, as this would result in serious vibration and pounding. Deep-well pumps are ordinarily built with too great a distance between bearing centers, which is sure to result in vibration and deterioration, and in decreased mechanical efficiency. The use of closely set bearings naturally adds to the cost of the machine, but only in this way can high mechanical efficiency and long life of bearings and shafts be secured.

21. The shaft **couplings** on Worthington AXIFLO pumps are made of the best grade of machinery steel, turned, bored, and threaded with long, straight left-hand threads. The shafts are correspondingly threaded at either end to screw into the couplings. The left-hand thread is to prevent the shafts from unscrewing with the standard counter-clockwise direction of rotation of the pump.

22. The question has arisen as to whether, when the pump is shut down and the water in the drop pipe passes down into the well and the impellers revolve as water wheels, the couplings will unscrew. They will not, for while the direction is opposite to normal, the power is being applied at the opposite end of the shaft and consequently the torque in the shaft is in the same direction under either condition of operation.

23. The impellers and diffusers or discharge vanes are all grouped at the bottom of the well in a single unit and they are so shipped. This relieves the customer of the responsibility and labor of **assembling** the pumping parts and it also concentrates all of the working parts except the shaft bearings at one point.

24. One more feature in the **barrel,** as it is called, warrants consideration. This is the method of attaching the impellers to the shaft. Each impeller is bored out to a standard taper and mounted on a steel compression sleeve tapered on the outside. The sleeve is made compressible by milling in, first from one end and then from the other, so as to permit it to clamp onto the shaft when the impeller is driven down over it. At the same time there is a hole drilled in the steel tapered sleeve and into the shaft, and by setting a round key in these holes the location of the impeller is absolutely fixed. This steel pin or key of large diameter is put through

the steel sleeve and into the hole in the shaft and then locked in place when the impeller is set down over the hole in the sleeve.

25. A final **lock for the impeller** is provided by a cylindrical nut screwed down over the top of the steel bushing and onto the top of the impeller and locked in place by an automobile-type copper star washer. In order to prevent any possible unbalanced condition of the impeller unit, the holes in the shaft for the keys are staggered so that the passageways of adjacent impellers are located at 180 degrees to each other.

26. The **impellers** are all carefully **finished** on their working faces and along the edges so as to be smooth and to pass through the water with a minimum of frictional loss, and they are also carefully balanced.

27. In contrast to this, the builders of cheaper types of deep-well pumps generally reduce the cost of manufacture of their axial flow or screw pumps by locating the impellers adjacent to the bearings at intervals throughout the length of the drop pipe, reducing the cost, of course, but involving a great deal of labor on the part of the purchaser to install. These impellers are crude in form, rough, unbalanced, and are in general held onto the shaft by means of set screws. The erector's judgment is relied on to see that the impellers are properly located and secured so that they will not move longitudinally on the shaft.

28. There is no engineering reason why the pump should not be used for wells deeper than 150 feet. It is rather a matter of commercialism. As each impeller unit has a comparatively low-stage head, the number of stages or impeller units will increase directly as the **head pumped against.** When the head materially exceeds 150 feet, the number of stages is so great as to increase the price to such an extent that either the Worthington CONIFLO (Reg. U. S. Pat. Off.) or the reciprocating type of pump becomes more attractive from a price viewpoint. Furthermore, either the CONIFLO or the reciprocating pump will show higher operating efficiencies at the higher heads.

29. In general the same limitations of capacity apply to the AXIFLO deep-well pump as apply to centrifugals. That is, the efficiency will be low where the quantities to be pumped are small—increased capacity means increased efficiency.

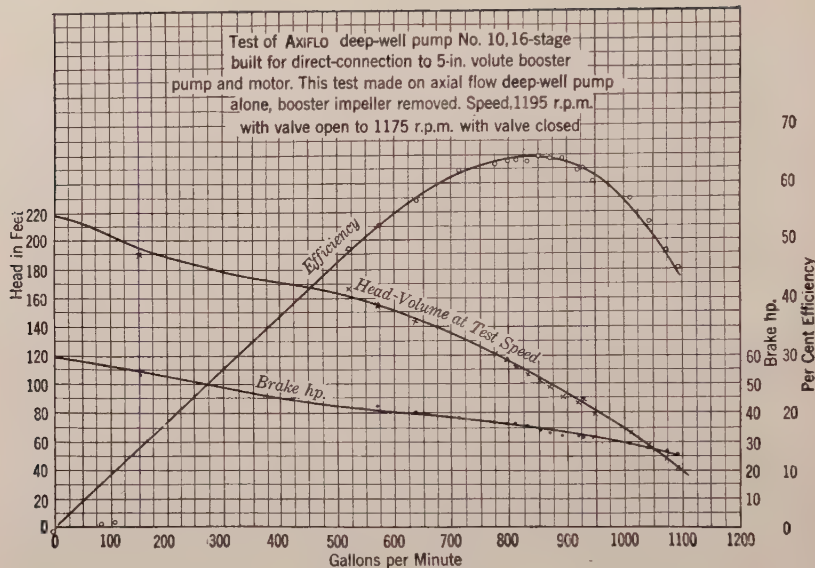


FIG. 10. Characteristic curves of No 10 16-stage Worthington AXIFLO Pump without Booster.

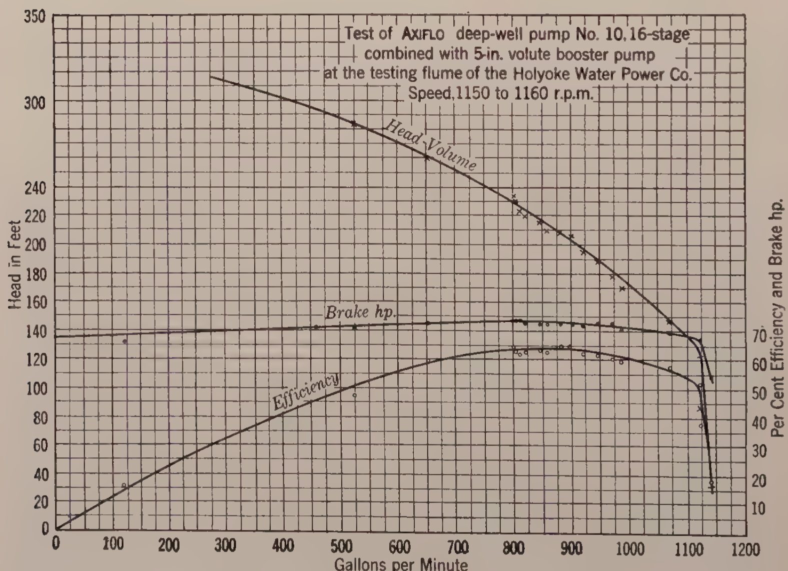


FIG. 11. Characteristic curves of same pump as shown in FIG. 10 but with Booster.

30. The **range of capacities** may be stated as follows:

Size Pump No.	Minimum Inside Diameter of Well—Inches	Capacity in Gal. Per Min. Min. to Max.
6	6	100 to 300
8	8	200 to 600
10	10	450 to 1200
12	12	800 to 2000
14	14	1200 to 3000
16	16	1800 to 4000
18	18	3000 to 6000

The very great majority of these pumps are driven by direct-connected vertical-shaft motors, and this being the case, the machines have been designed to operate at the full-load **speeds** of 60-cycle motors; namely, 1750 r.p.m. for the No. 6 and No. 8 pumps; 1150 r.p.m. for the No. 10, No. 12, No. 14 and No. 16 pumps; and 860 r.p.m. to 1150 r.p.m. for the larger machines. The speeds may, however, be modified, for 50 cycles, 40 cycles, or for 25 cycles.

31. Characteristic curve, Fig. 10, was taken from a No. 10 16-stage pump tested at the Holyoke Water Power Company's flume, while curve Fig. 11 was taken from a similar size pump with booster attached. The head-capacity and efficiency curves are especially interesting. It is to be noted that the head-capacity curve bears considerable resemblance to the corresponding curve of a centrifugal pump, and that the efficiency curve is very flat, which insures maximum efficiency throughout a considerable range of heads and capacities.

32. On this type of pump there is no possibility of overloading the motor on account of a decrease due to the pumping water-level being higher than anticipated, as will be noted from the brake-horsepower curve. On the combined AXIFLO pump and booster units an almost ideal condition exists, the brake-horsepower curve being almost flat throughout the entire range of the pump's operation.

33. The outstanding **advantages** of the Axiflo pump are:

1. Has a capacity from three to six times that of a reciprocating two plunger-pump of the same well diameter.
2. Will pump an average of 100 per cent more water than a centrifugal pump from a given size well.
3. Can be used in wells too small for a centrifugal pump, as well as in those of large diameter.
4. Has but a few simple parts (no valves); hence a long life.
5. Is easy to operate.
6. Discharges a continuous, uniform stream of water, producing steady load on driver.
7. Combines in one unit, when desired, both a deep-well pump and a booster pump.
8. Cannot overload driver, even if capacity is increased by decreasing head.
9. Operates quietly, no pulsations.
10. Does not need priming, because lowest impeller is always submerged.
11. Occupies little floor space.
12. Low cost of complete plant due to smaller diameter well required for a specified delivery.
13. Suitable for either belt drive from oil engine or other prime mover or direct connection to standard vertical motor.

34. Field of Application.—The great field for the Axiflo pump is in wells 6 in. or more inside diameter of casing where large volumes of water are available and where the level when pumping does not recede more than 100 to 150 ft. below the ground surface.

35. The Axiflo pumps are in use in small and moderate sized **water works,** in replacement of reciprocating pumps and particularly of air lifts. Remarkable reports of increased water supply and reduced pumping cost on installations of this character have been received. They are also being used for **irrigation, ice manufacturing** plants, **creameries, paper** and other **mill** supplies, in shallow mines, **clay pits,** and in **coal pits.**

36. There is an installation in an **ice cream factory** in Springfield, Mass., which has been in service now for over five years which is typical of what may be expected. At this place an Axiflo pump

replaced a reciprocating pump, which it was thought was supplying all of the water obtainable from an old well. A No. 8 Axiflo placed about 110 feet below the surface discharges approximately 235 gallons per minute as against about 125 of the old type of machine.

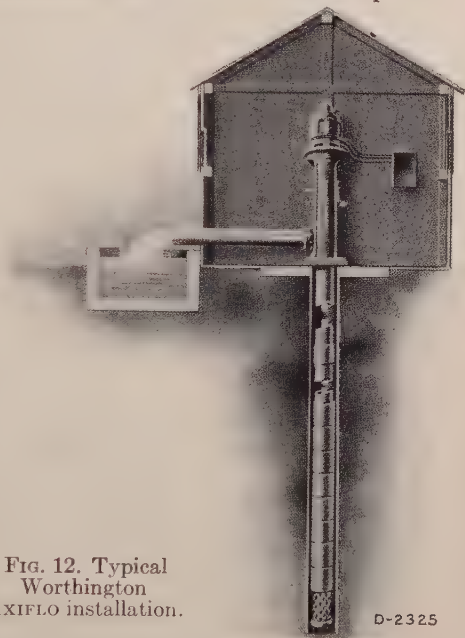


Fig. 12. Typical Worthington Axiflo installation.

The water runs at a uniform temperature of 52 degrees to 54 degrees Fahrenheit, and 235 gallons per minute is sufficient to run the plant without the use of city water, which during the summer has a temperature of approximately 66 degrees to 67 degrees Fahrenheit. The reduction in temperature of the **cooling water** supply has resulted in a net saving of 25 per cent in the total plant operating cost, and this is a reasonable estimate of the probable saving in any plant of this kind where sufficient well water is available.

37. There is a place for an Axiflo pump in most wells where an **air lift** is now installed, and in almost every instance the Axiflo pump will pay for itself in a very short time.

WORTHINGTON **CONIFLO** PUMP

(Reg. U. S. Pat. Off.)

38. For pumping wells from approximately 150 to 400 ft. in depth, the Worthington Coniflo Pump is highly recommended. From a well of any given size the Worthington Coniflo Pump will deliver more water, develop a greater head per stage and operate at a higher efficiency than any centrifugal deep-well pump.

39. The **characteristic curve** of a No. 20 Coniflo pump is shown by Fig. 13. The steep "head capacity" curve indicates a considerable range in head with a small percentage of variation in capacity. The water level in the well may rise or sink within wide limits

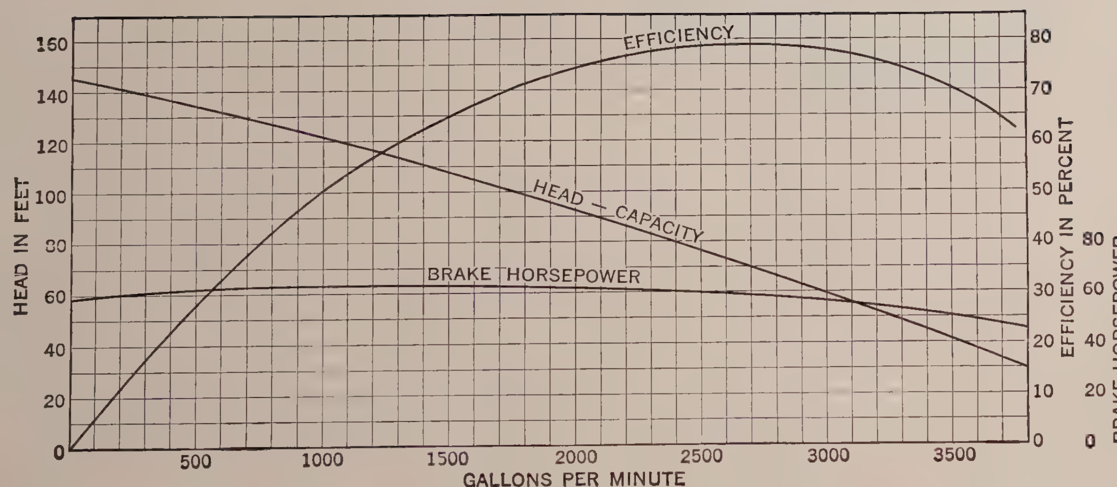


Fig. 13.

Characteristic Curves of No. 20 Worthington Coniflo Pump.

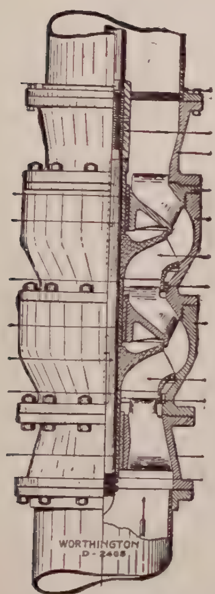


FIG. 14. Half section showing details of impeller unit of Worthington CONIFLO Pump.

without materially affecting the capacity of the CONIFLO pump. The efficiency of the CONIFLO is not only high when the pump is operating under the conditions for which it was designed, but the curve also indicates a very slight drop under widely varying conditions of head and capacity. The horsepower curve is almost flat, showing the impossibility of overloading the motor under changing conditions.

40. The **driving heads, thrust** and **guide bearings** of the CONIFLO pumps are practically the same as the AXIFLO described in the preceding pages.

41. The impeller or pumping unit of the Worthington CONIFLO Pump is shown by Fig. 14. The number of impellers may be from one to twenty or more, depending upon the total head the pump is to work against.

42. The **impeller**, Fig. 15, is made in one piece and is smooth finished all over. The vanes are designed to utilize the axial-flow principle modified by a semi-radial discharge in order to secure the high head per stage and the other desirable features shown by the characteristic curve Fig. 13.

FIG. 15. Impeller used in Worthington CONIFLO Pumps.

43. Guide vanes above each impeller guide the water from the discharge of one impeller to the suction of the next and prevent eddy currents with their resultant hydraulic losses. Each guide vane is cast integral with its individual section of the pump casing. The guide vanes also support the shaft bearing.

44. Above and below each impeller is a phosphor-bronze sleeve

which serves the double purpose of a shaft bearing and a spacer to correctly locate each impeller on the shaft.

45. The Worthington CONIFLO Pump is built in the following sizes having capacities from 200 to 3500 gal. per min:

Size of Pump No.	Capacity Gal. per Min.	*Size of Well Casing—Inches
12	200 to 1200	12
14	400 to 1800	14
16	600 to 2200	16
18	800 to 2600	18
20	1000 to 3500	20

*Minimum inside diameter in inches.

RECIPROCATING DEEP-WELL PUMPS

46. For depths greater than that for which the CONIFLO Pump is suited, Worthington offers several types of reciprocating pumps. The **GLENDORA** (Reg. U. S. Pat. Off.) triple-plunger pump, Fig. 16, has a larger capacity than any other type of reciprocating deep-well pump, is suited for extreme depths, and is of high mechanical efficiency.

47. The GLENDORA Pump gives approximately three times the volume of water that can be obtained with a single-plunger, single-acting pump of the same size and approximately fifty per cent more than either the double-acting, single-plunger, pump or the two-plunger pump.

48. The load is as steady as in a triplex pump. The plungers alternately coming into action at approximately their maximum speed and being so timed that one takes up the full load as the preceding one relinquishes it, insures a constant pressure on the water column and a uniform load on the driving mechanism. The steady flow adapts the pump especially to service where it is desired to pump directly into the distributing system.

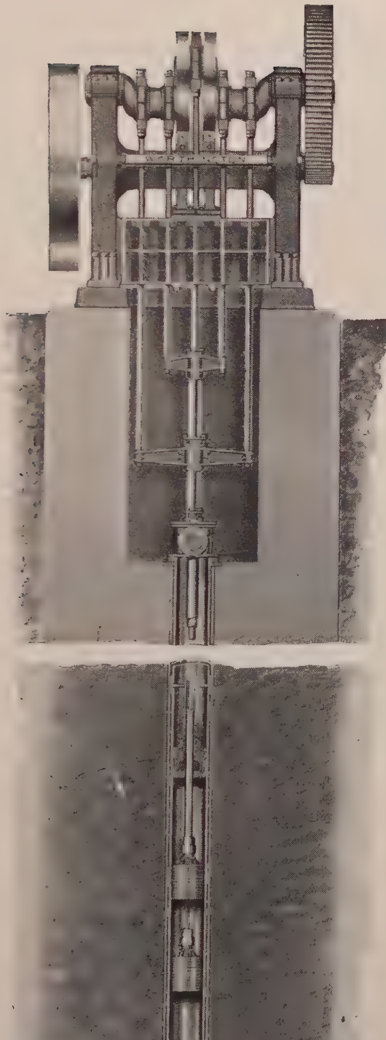


FIG. 16.

WORTHINGTON GLENDORA
DEEP-WELL PUMP.

SECTION IV

Power for Pump Drives

(Figures refer to paragraph numbers.)

Oil-engine drive, 1–8; Gasoline and kerosene engines, 15–16; Waterwheel-driven pumps, 20–36; Steam turbines, 40–48; Electric motors, 50–89; Units of electrical measurements, 90-106; Electrical equivalents, 107; Full-load speeds for a.c. motors, 108; Ampere ratings of a.c. and d.c. motors at full load, 109; Approximate amperes per terminal for induction motors, 112; Copper-wire table, 113; Carrying capacity of cables, 114.

SECTION IV

Power For Pump Drives

1. **Oil-Engine Drive.**—The oil engine, due to its low standby charges as well as its low operating costs, is desirable as a driver for pumps of either the positive displacement or centrifugal type.

2. With oil engines, it is important to keep the load at less than the 100 per cent rated horsepower of the engine. To meet a reduction in the **power** required by a pump by a reduction in the **speed** of an oil engine has the unfavorable effect of reducing the power available in the engine in the ratio of speed reduction. The fact that the power drops as the cube of speed reduction in a pump, and as the first power of speed reduction in an engine, affords some measure of relief, but the field of application is small.

3. An oil engine should be so selected that in the event it has to work over a head range its **power demand** should not at any point exceed 90 per cent of its rated power. If speed reduction on an oil engine is involved, the power demand at reduced speed should not exceed 90 per cent of the mean effective cylinder pressure at that speed.

4. In some of the low-head drainage units, it has been found very practical to cut the speed governor out of action and use it entirely for emergency speed control and to **regulate the engine** by a device which limits the amount of oil injected per stroke. In other words, we maintain the mean effective pressure in the engine cylinder and allow the speed to follow the demand for power according to the head variation on the pump.

5. The **screw-impeller type centrifugal pump** has a decided advantage over the pumps of other types **for oil-engine drive.** As the head on a screw pump is reduced, the power is also reduced and the engine is so much underloaded as to permit of raising its speed somewhat and obtaining greater capacity from the pump. This is the opposite from the ordinary centrifugal pump.

6. The **field of the oil engine** for driving centrifugal pumps is not confined to drainage and low-head irrigation work where the medium-speed drainage or the screw pump is employed. By the

use of suitable speed-up gears, the most favorable pump speeds can be obtained for high-head service. Already a large number of Worthington pumps geared to Worthington Diesel engines are doing excellent and continuous duty in municipal water works.

7. Worthington is fully prepared to furnish complete pumping units consisting of both oil engine and pump of their own manu-

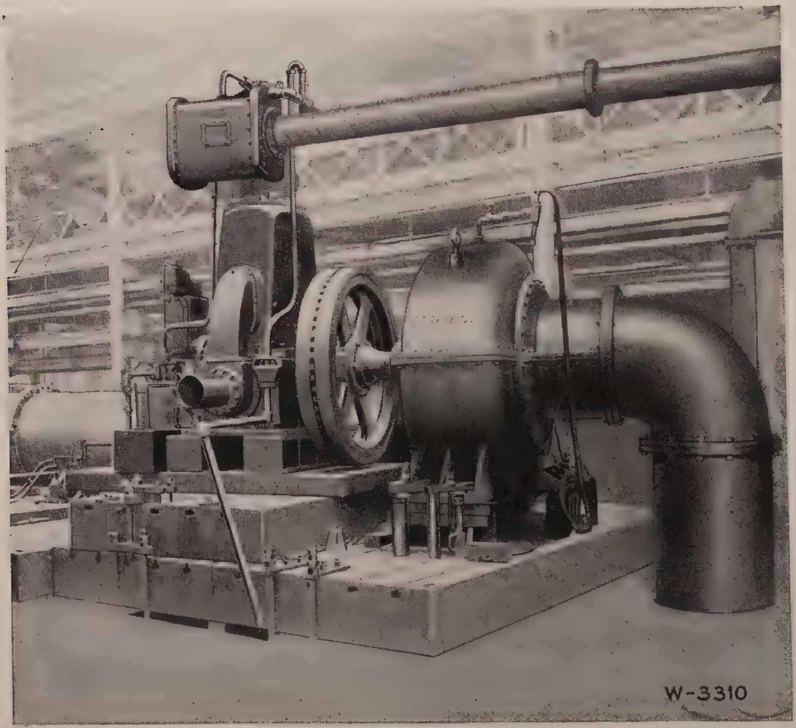


Fig. 1. Worthington 75-hp. Single-cylinder Two-cycle Solid-injection Oil Engine connected to drainage pump running at 275 r.p.m. and delivering 19,000 gal. per min. at 10-ft. head.

facture, relieving the purchaser of any possible annoyance or delays arising from divided responsibility for engine and pump.

8. Fig. 1 shows drainage unit, consisting of a Worthington medium-speed drainage pump for 19,000 gal. per min. at 10-ft. head, direct-connected to a 75-hp. 275-r.p.m. Worthington Oil Engine.

15. Gasoline and Kerosene Engines.—Stationary gasoline and kerosene engines are frequently used to drive centrifugal pumps by means of belts. An installation of this kind for a small irrigation project is shown by Fig. 2.

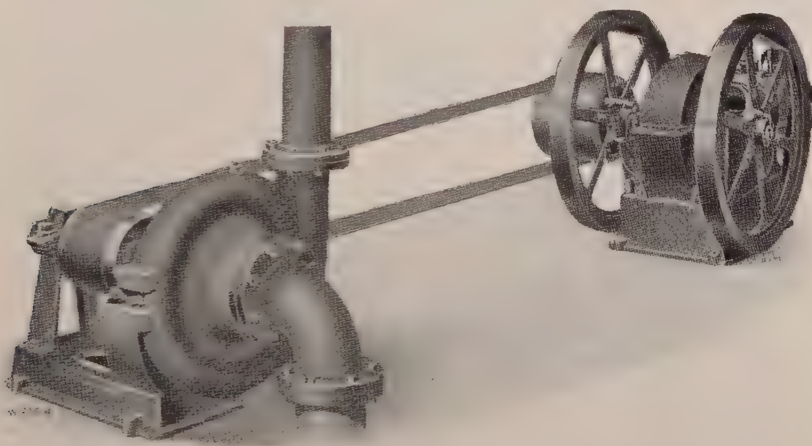


Fig. 2. Kerosene engine belted to centrifugal pump.

16. The high-speed marine-type gasoline engine is sometimes direct-connected to fire pumps, or other pumps installed for emergency service. Fig. 3 shows an 8-in. Class SD Volute Pump direct-connected to a 200-hp. eight-cylinder marine-type gasoline

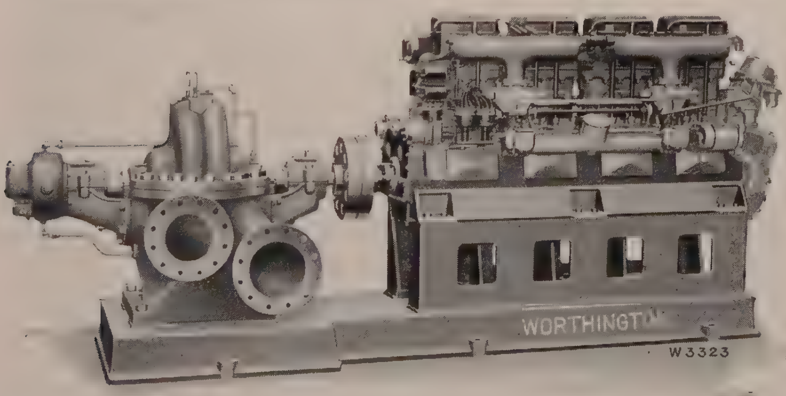


Fig. 3. 200-hp. high-speed marine-type gasoline engine connected directly to 8-inch Worthington Class SD Volute Pump.

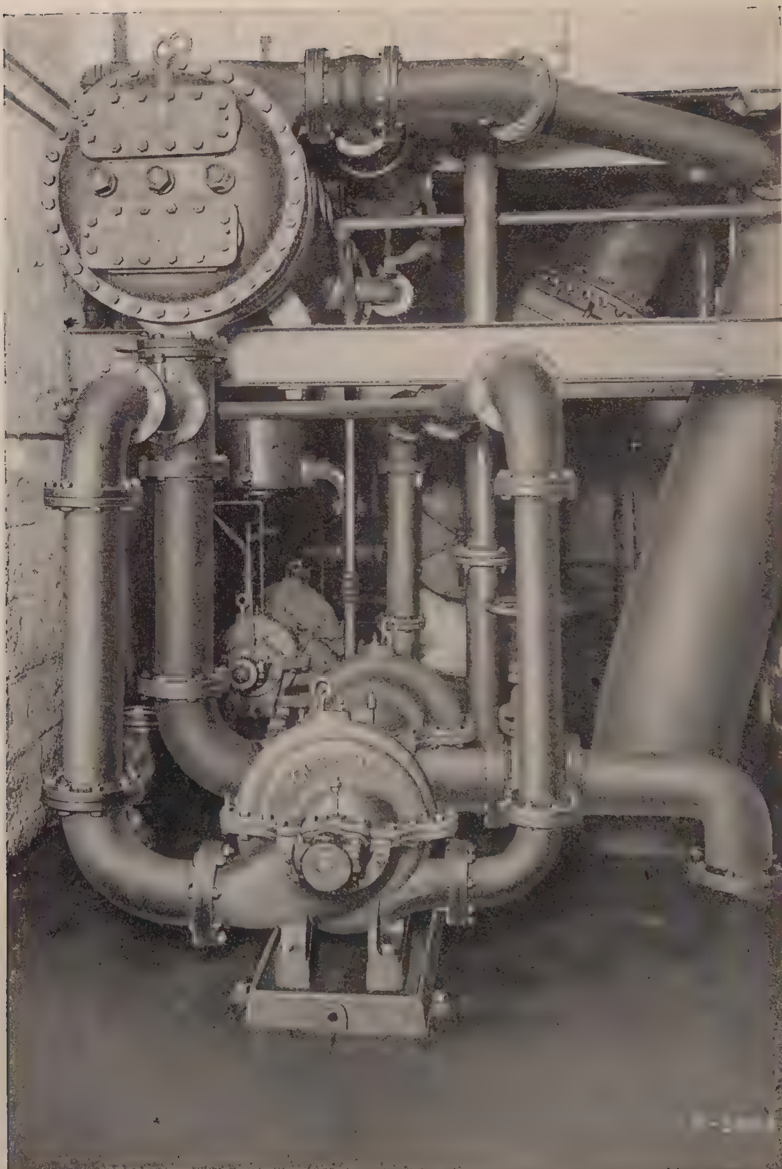


Fig. 4.

8-Inch Class OS Worthington Circulating Pump Driven By 4-Inch Inlet Water Wheel, Also a 2-Inch Class SD Two-Stage Condensate Pump Driven By 4-Inch Inlet Water Wheel.

engine. Operating costs prohibit the use of the large gasoline engine for continuous service where other sources of power are available.

20. Water-Wheel Driven Pumps.—Water wheels make a very attractive drive for centrifugal or power pumps as there is always a reasonable amount of flexibility in the speed, which obviously is a desirable feature.

21. Worthington has developed and built a number of these combinations, using reaction turbines both of the fixed-vane and wicket-gate type.

22. In considering this type of drive, it must be remembered that it is not always necessary to have a natural water power available. There are a number of installations where pumping is necessary, yet where it is not desirable to install electric motors or other types of drive. In such cases, it is customary to generate water pressure by a central pumping plant and to use this energy for driving one or more small fixed-vane power turbines, either belted or direct-connected to centrifugal or reciprocating pumps.

23. A system of this kind can be of the continuous circulating type, in which the water passing through the power turbine can discharge under a back pressure to a header and be returned to the sump of the pump which generates the pressure. This feature of discharging against a back pressure can only be accomplished with a reaction turbine, as an impulse turbine requires free atmospheric discharge. With the reaction wheel one or more turbines may be operated in series where the pressure head is such as to permit it.

24. Fig. 4 shows an 8-in. class OS circulating pump driven by a 4-in. inlet water wheel, also a 2-in. class SD two-stage condensate pump, also driven by a 4-in. inlet water wheel.

25. Water wheels are now used quite generally for driving the auxiliaries for steam turbine-driven centrifugal pumping units for water-works service. With hot-well and circulating pumps driven in this manner, flexibility is obtained, and the resultant duty of the main unit is usually increased over other methods of drive. Pressure water is taken from the main pump and discharged back into the suction without wasting water, thus obtaining auxiliary energy at the low water rate of the main steam turbine.

26. Another **water-works application** is for driving filtration plant wash-water pumps. The water required for driving the hydraulic

turbine is taken from the discharge main leading from the main pumping engine. After passing through the hydraulic turbine, the water is discharged into the overhead wash-water supply tank. The amount of water to be handled by the wash-water pump can thus be reduced by the amount of water that is required to drive the hydraulic turbine. This method as a rule is more economical, flexible and controllable than electric motor drive.

27. For **booster work,** and general low-head pumping, this type of drive is sometimes applicable. Fig. 5 shows such an installation consisting of a 65-hp. Worthington wicket-gate hand-operated

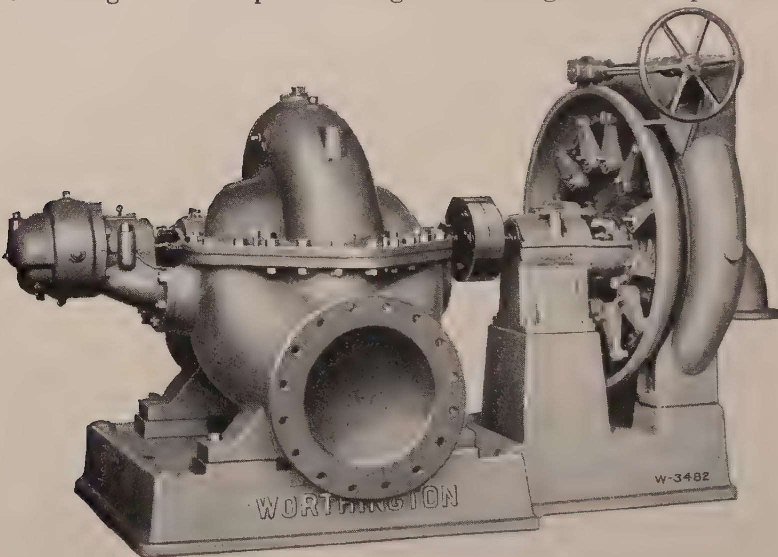


Fig. 5. Hydraulic-turbine-driven Centrifugal Pumps.

reaction turbine, operating under a head of 175 ft. at a speed of 1000 r.p.m. and direct-connected to a Worthington 16-in. Type OS pump delivering 6950 gal. per min. against a head of 25 ft.

28. Water-wheel drive can be used to advantage on **irrigation** projects where a gravity system exists and part of the water is required at a high elevation. The pump in this case is a booster receiving water under pressure from the gravity line, while the water for driving the wheel, after giving up its energy, passes on down to lower levels that need irrigation. Such an installation is shown by Fig. 6. Two of these units were furnished, each con-

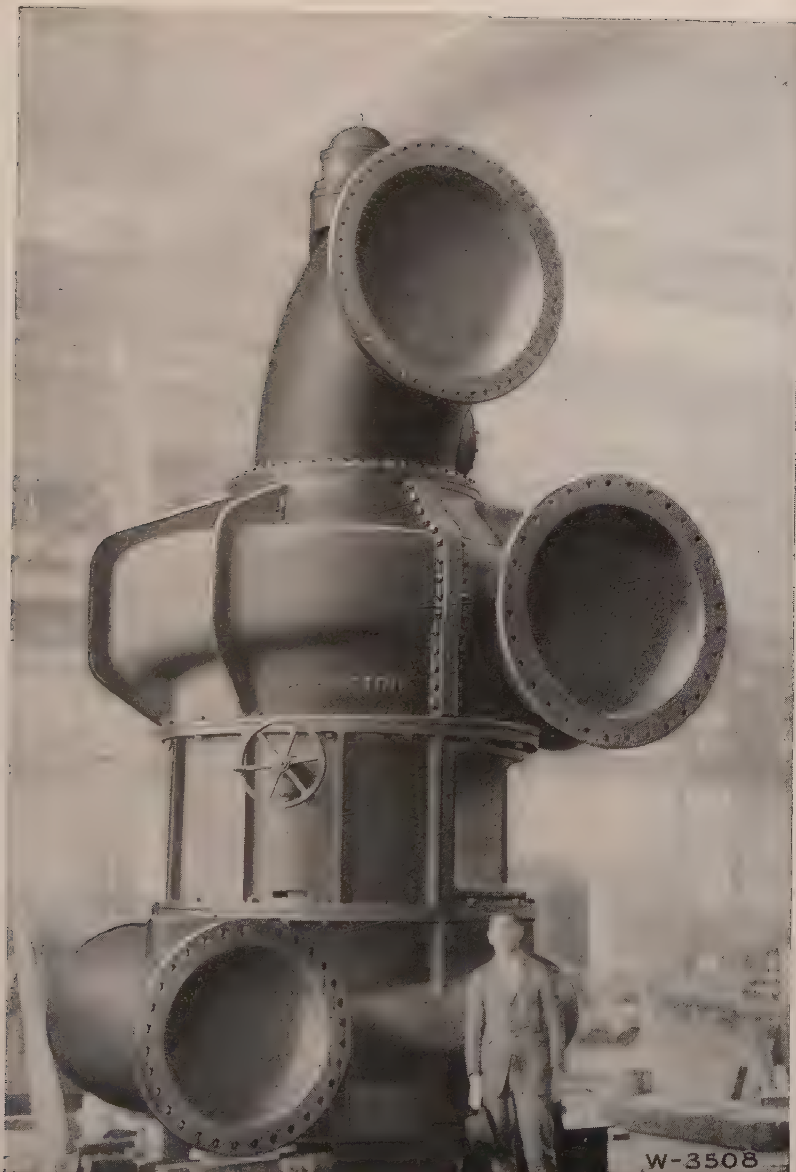


Fig. 6.

Irrigation Pump Driven By A Water Wheel. Water Used For Driving Employed For Irrigation After Its Energy Has Been Extracted. 550-hp. Vertical Screw Pump Delivering 60,000 Gal. Per Min. At 28.5-ft. Head At 380 r.p.m.

sisting of 550-hp. Worthington Vertical Turbine operating under 89 ft.-head, direct-connected to 48-in. Worthington Vertical Screw Pump, delivering 60,000 gal. per min. at 28.5-ft. head at a speed of 380 r.p.m.

29. Water-turbine application can be worked to advantage on **pumping water for cities** where there is not sufficient natural power head to force the water through long discharge lines, but where the volume of natural power water is sufficiently in excess to drive a turbine, which in turn is connected to power or high-pressure centrifugal pumps.

30. A number of **central power stations** are now considering the use of turbines for driving pumps or other mechanical devices in stations where it is desirable to eliminate either electric wiring or steam lines and their corresponding types of drives.

31. Other combinations of this drive are: Driving pumps on evaporator work, such as **sugar house** and **salt works**; as **safety units** for driving pressure pumps; for **mining work** in case of flood conditions; or operating cutters and shifting apparatus on Diesel engine-driven dredges.

32. In order, however, to determine whether water-wheel drives are suitable, it is necessary to know the pumping conditions which determine the type of pump, drive, speed and horsepower necessary to be provided in the water wheel. In cases where manufactured power is to be considered, it is, of course, rather easy to **determine the size of water wheel** after knowing the pumping conditions, but in order to determine whether there is sufficient power available on the natural water supply, the following relation should be used:

$$F = \frac{P \times 8.82}{h \times \mu}$$

Where F is the flow of water in cubic feet per second; P the brake-horsepower; h the net head in feet and μ the efficiency in per cent. Roughly, for 80 per cent turbine efficiency the brake-horsepower is

$$P = \frac{F \times h}{11}$$

33. The maximum speeds permissible on water wheels are dependent on the power head, and are usually expressed in terms of specific speed. The specific speed forms the basis of classification

and may be defined as the speed in revolutions per minute of a runner if reduced in all dimensions so that it would develop one horsepower under one-foot head.

34. The following simple relation can be used in order to **determine specific speed** of wheel (not to be confused with specific speed for centrifugal pumps):

$$S = \frac{N}{\sqrt{h}} \times \sqrt{\frac{P}{h \times \sqrt{h}}}$$

Where S is the specific speed; N the revolutions per minute; P the brake-horsepower at full gate, and h the net head in feet.

35. The **maximum specific speeds** that are permissible for reaction wheels for various heads may be stated approximately as follows:

Head	Specific Speed
25	110
50	100
75	70
100	55
200	40
300	32
400	30

36. From the above, the maximum turbine speeds can be determined for a given horsepower and head. It is, of course, permissible to use lower speeds than calculated from the above, to meet pump requirements.

40. **Steam Turbine.**—The steam turbine and the centrifugal pump are both high-speed machines. They are very well suited to each other, and combine to form an efficient, flexible, compact unit. With the recent developments in the small sizes of steam turbines, this form of driver has become quite economical.

41. The **rating of steam turbines** in connection with centrifugal pumps should always be on the maximum load, otherwise the expected efficiency will not be obtained. The steam turbine also gives a greater factor of safety for overloads. Should an **overload** condition occur, a motor of the 50-deg. type could be damaged if improperly selected. Under the same conditions a steam turbine

would simply reduce its speed to a point where the power developed by the turbine would equal the power demand of the pump.

42. The steam turbine-driven pump is more flexible than the motor-driven unit, as the speed of the turbine can be easily altered by adjusting the governor. The turbine can be equipped with a governor operated by the discharge pressure of the pump, in which case the **speed** of the turbine is **automatically regulated.** This is of great value in boiler-feed service as the discharge pressure produced by the pump is constant, regardless of whether the pump is operating at full, three-quarter or half load.

43. On account of its reliability and fewer parts, a simple single-stage turbine is usually employed for boiler-feed and general service pumps. A direct-connected centrifugal type of governor, with an emergency over-speed stop, is all that is needed to **regulate the turbine.**

44. On boiler-feed work, where a pressure regulator is used to maintain pressure at a fixed value, the speed governor does not come into use at all, unless the pressure governor tends to make the turbine run at excessive speeds to supply excess capacity. This excess capacity requires greater horsepower and the turbine probably could not develop such excess power, so that a speed governor really has no use in boiler-feed service. Nevertheless, an emergency speed governor should always be employed if the speed governor is left off the turbine when a pressure regulator is used. The pump may become vapor bound and lose its pressure, in which event the regulator would open and the steam supply be increased and the machine run away unless checked by either the speed governor or the **emergency speed governor.**

45. Small steam turbines for driving pumps are usually operated non-condensing. Small and medium size pumps for moderate or high heads may be direct-connected to the turbine. A reduction gear, however, must be used with the larger pumps to enable both the pump and turbine to operate at their most economical speeds. The larger turbines are generally operated condensing. In water-works service, the condenser is placed in either the suction or discharge line, and the water pumped by the main unit is utilized as cooling water in the condenser.

46. The **type of steam turbine** is governed to a certain extent by

the size of the pump and the service. The large turbines for **water-works service** are of the multi-stage condensing type or the multi-stage bleeder type or a combination of the two. The steam turbine has proved to be an efficient and economical driver for centrifugal pumps for water-works service which demand a pump for low-pressure domestic and high-pressure fire service. The turbine is designed with additional steam nozzles that can be cut in for fire service without affecting the economy of the turbine for domestic service.

47. For **domestic service,** the operation of the unit is controlled by the pressure and speed governors, so that a constant pressure is maintained for full or fractional loads. For **fire service,** the pressure line to the governor is closed, and the additional steam nozzles are opened, speeding up the turbine with a corresponding increase in the discharge pressure of the pump.

48. In **steel mills,** where there is an abundance of exhaust steam from the rolling-mill engines and hammers, the mixed-flow type can be used to advantage.

50. Electric Motor.—The most common and the simplest driver for centrifugal pumps and for power-driven displacement pumps is the electric motor. To obtain the best results, it is essential that the **motor selected** for driving the pump be of the proper size and type. The capacity and rating of the motor are of great importance, especially when power is purchased on the rating of the motor. If the motor is too small, it will be constantly overloaded, and if too large, the customer pays for power not used. Local conditions may govern to some extent the type of motor and controlling equipment to use for some specific installation. In the majority of cases, however, the selection will be governed by the characteristics of the pump, the variations in load during operation, the characteristics of the motor and the characteristics of the pump and motor as a whole.

51. All motors used for driving pumps should have a high grade of moisture-resisting **insulation.** Where excessive moisture is encountered, as in deep, poorly ventilated pump pits, damp mines, etc., a special moisture-resisting insulation should be specified.

52. Motors for alternating-current installations are of several types, each possessing certain features that render it best for some particular service condition.

53. Electrically and mechanically, the **synchronous motor** is the same as the alternating-current generator. Such motors are called synchronous because they always run in synchronism (i.e., with the same frequency) with the generator supplying the current to them. The synchronous motor is a constant-speed motor.

54. The speed at which it will run can be calculated from the formula

$$N = \frac{2 f\, 60}{p}$$

where N is the speed of motor in revolutions per minute; f the frequency of the alternating-current supply, and p the number of poles on the motor field.

55. The synchronous motor, especially in large units, possesses a number of features which make its use at times preferable to that of the induction motor. Its **advantages** may be briefly summed up as follows:

1. Unvarying speed at all loads.
2. Power-factor variable at will by change of the exciting current, can be made approximately unity at any load.
3. In general, induction motors are cheaper than synchronous motors, except on the larger slow-speed units, where the synchronous motors are often cheaper.
4. Its efficiency is generally higher than that of the induction motor.
5. It is especially adapted to high-voltage winding.
6. When used in combination with inductive loads, the synchronous motor will improve the electrical efficiency of the system, since it can be built to operate at a leading power-factor (leading magnetizing current) so as to counterbalance in whole, or in part, lagging magnetizing currents taken by induction motors.

56. The synchronous motor, on the other hand, has several **disadvantages,** as follows:

1. It is not adapted to work requiring variable speed, as no independent speed regulation is possible.
2. The standard line of synchronous motors is designed for a starting torque of 50 per cent, pull-in torque of 50 per cent with 70 per cent voltage applied. Greater pull-in torque will be obtained by applying full voltage with the field switch open.

3. On a centrifugal-pump load, a well designed synchronous motor will not hunt. When applied to a reciprocating pump, however, great care must be used, for unless the design of the motor is carefully checked, it may tend to oscillate, causing pulsations injurious to the motor.
4. It requires an exciting current which must be supplied from an outside source.
5. It requires the most skillful and intelligent attention.

57. Analyzing the advantages and disadvantages, we find that, due to the benefits of **power-factor correction,** synchronous motors are often permitted where squirrel-cage motors are objectionable, even though the line disturbance at starting may be the same. Pump units of 75-hp. or larger, which operate continuously, afford an ideal opportunity for economically improving the power-factor.

58. For **slow-speed** centrifugal pumps, synchronous motors are doubly desirable from the standpoint of both power-factor and first cost. Before purchasing a synchronous motor for driving a pump, the operating conditions must be carefully analyzed in order to determine if this type of motor is suited for the service. Unless all phases of the pump and motor characteristics are carefully investigated, one cannot be assured a successful installation.

59. The **synchronous motor in starting** must attain synchronous speed before it will lock into electrical step with the incoming current. To do this the motor must have characteristics that enable it to generate sufficient torque through the last 2 or 3 per cent of speed range to attain synchronism. This torque during the electrical locking-in period is the **pull-in torque,** and unless it equals the power demand from the pump the motor cannot attain synchronism.

60. We will take, for **example,** a centrifugal pump of the single-stage volute type. A pump of this type when driven by a synchronous motor must be started with the discharge valve closed, since by so doing the pull-in torque is decreased. By referring to the comparison of runner types, (Sec. II, Par. 21, Fig. 5), we find that the horsepower for starting at shut-off zero capacity varies from 42 per cent for the lowest speed runner to 105 per

cent for the mixed flow runner. This horsepower range also represents the pull-in torque range required of the motors for driving.

61. The standard synchronous motor, in speeds above 900 r.p.m., develops only about 60 per cent pull-in torque. Lower speed motors have lower pull-in torque. For a 360-r.p.m. motor it is about 42 per cent. Therefore, a standard synchronous motor could be used for driving a constant-speed centrifugal pump with a runner having the characteristics of Case 1*. For a pump with a runner having the characteristics of Case 2* or Case 3*, a special synchronous motor must be used, and in cases such as these the problem should be referred to the manufacturers as to the pump requirements and as to what pull-in torque the motor can produce.

62. The **super-synchronous motor** is especially well adapted for pump loads requiring especially high starting and pull-in torques. A special mechanical feature of this motor is that the stator can rotate during the starting period. When starting with load, the armature (stator) is energized and comes up to induction-motor speed. The field is then energized and the armature (stator) comes up to synchronous speed. By applying the band brake with which the stator of this machine is equipped, the load is accelerated to synchronism proportionately as the armature is brought to rest. The motor, therefore, has torque up to the breakdown of the motor available for starting and accelerating the load.

63. The **Fynn-Weichsel motor** is a general-purpose motor that combines the operating characteristics of the synchronous motor described above and the slip-ring induction motor described below. It is made in sizes from 5 hp. 1800 r.p.m. to and including 200 hp. 720 r.p.m. for 60-cycle alternating current, with four- and six-pole ratings in the small frames and six-, eight- and ten-pole ratings in the larger frames and twelve-pole ratings in the largest sizes.

64. It starts as a slip-ring induction motor, having starting torque characteristics much more favorable than the usual squirrel-cage motor; it will develop 150 per cent torque with a starting current of from 150 to 200 per cent normal current. It requires no hand regulation of its direct-current field, as this regulation is inherent in the design of the motor. After the Fynn-Weichsel motor obtains synchronous speed, which it will easily do with 150 per cent load, it becomes a self-excited synchronous motor. As a syn-

*See Sec. II, Par. 21, Fig. 5 and Fig. 5A.

chronous motor it will carry 150 per cent load without falling out of step. Loading a Fynn-Weichsel motor beyond the limit of synchronous-speed operation simply causes it to act as an induction motor and it will immediately pull back into step when the overload is reduced. It will carry up to 300 per cent rating without stalling.

65. Advantages:

1. When used in combination with induction motors, its leading magnetizing current will counterbalance a lagging magnetizing current of equal amount taken by induction motors. If a Fynn-Weichsel motor is paired with an induction motor of the same horsepower and speed, the power-factor of the combined load will be substantially unity, irrespective of the loads on the two types of motors.
2. Unvarying speed for loads up to 150 per cent rating.
3. Heavy starting torque without excessive starting current.
4. Pull-in torque equal to starting torque.
5. Overload capacity equal to that of the induction motor.

Disadvantages:

1. Not adapted to work requiring variable speed.
2. Its cost is higher than induction motors.
3. Maintenance is equal to that of direct-current motors.

66. Induction motors are commonly divided into two types, the squirrel-cage and the slip-ring. When an induction motor is running without load, its speed is nearly equal to the speed of the rotary magnetic field; namely, synchronous speed. When the motor is loaded, its speed decreases to about 98 per cent of the synchronous speed in the case of large motors, and to about 92 per cent of the synchronous speed in small motors, at full load. The decrease in speed expressed as a percentage of synchronous speed is called the "slip" of the motor.

67. The **squirrel-cage motor** derives its name from its rotor winding which consists of heavy copper bars which in motors of the best construction are either cast integral with the end rings or welded to these rings, forming a structure similar to the exercising wheel on the cages used for pet squirrels; hence the name "squirrel-cage." The squirrel-cage motor requires from three to four times full-load **current** to produce at **starting** a torque equal to the torque developed when running at full load.

When the motor has to start under a heavy load, or where the taking of excessive currents from the supply mains will interfere with other apparatus supplied from the same mains by causing excessive drop in voltage, the squirrel-cage motor is objectionable, especially in large-size motors.

68. The extreme **simplicity of the squirrel-cage motor,** and its ability to carry enormous currents without injury, largely compensate for these disadvantages. A squirrel-cage motor will develop sufficient torque to start satisfactorily with from 40 per cent to 60 per cent of the rated voltage applied to the primary member. Therefore, the current required for starting may be greatly reduced by supplying the primary member at starting with current through a step-down transformer which is designed to reduce the supply voltage to a variable percentage of the rated voltage of the motor, and to multiply the delivered current in the same ratio. This step-down transformer is usually an **auto transformer** and, with its special switching device for changing the motor connections quickly from the low starting voltage to the full running voltage, is called an auto-starter, or a compensator.

69. **Standard compensators** for sizes up to 18 hp. are provided with taps to give 50, 65 and 80 per cent of line voltage; for larger motors, taps are provided to give 40, 58, 70 and 85 per cent of line voltage.

70. The **slip-ring motor** also derives its name from the construction of its rotor, which is wound with insulated wire. The terminals of this winding are connected to a starting resistance mounted external to the motor. This resistance may be designed for starting duty only or for speed regulation. The slip-ring motor takes at **starting** only about full-load rated **current** from the supply mains, giving a starting torque about equal to full-load torque. This type of motor is used only where a starting torque, not greatly in excess of full-load torque, is required. This motor has the advantage of not taking excessive current at starting, and will start, therefore, without producing excessive drop of voltage in the system from which the motor receives its power. As the motor increases in speed the resistance is cut out in as many successive steps as there are contact points, allowing the pump to come up to speed more quickly and with more uniform accelera-

tion than is possible with the squirrel-cage motor. This is a very desirable feature in centrifugal pump operation. Large-size motors are always of the slip-ring type except in mine installations.

71. Alternating-current motors have been developed of the adjustable varying speed **commutator type,** which eliminates the undesirable secondary resistance losses and which possesses high efficiency at reduced speeds. This type of motor is seldom required for driving centrifugal pumps but is sometimes desirable for driving positive displacement pumps. These commutator-type motors should be considered for applications where the pumps run at greatly reduced speeds for considerable periods, as the efficiency and power-factor are both higher than those of the corresponding slip-ring induction motors under these conditions and a considerable saving in power will result by the use of commutator motors. This type of motor has the "wound" rotor with the terminals brought out to collector rings on the shaft. The brushes are shifted by means of a handwheel or other device. These motors have excellent starting characteristics and are capable of 50 per cent speed reduction.

72. Direct-current motors are of three kinds—series, shunt and compound—and as in the case of the alternating-current motor, derive their name from the winding. In **series motors,** the speed varies with the load. If part of the load is taken off, the fields are weakened, causing considerable increase in the speed. If the load is entirely removed, the speed becomes dangerously high. This type of motor is used where a constant speed is not necessary and where the load cannot be entirely removed. The series motor is not used for driving pumps.

73. The **shunt motor** is a constant-speed motor. It is used extensively for driving pumps where the head is practically constant, and for intermittent service.

74. The **compound motor** is practically a constant-speed motor capable of developing a heavy torque on starting. This type of motor is used wherever there are variable horsepower demands with little change in speed. For driving pumps compound-wound motors are generally used. For constant-speed service their starting characteristics are better than the shunt-wound

motor. For adjustable-speed service the compound-wound motor is always used.

75. The direct-current motor does not attain constant speed until it has run long enough to get thoroughly warmed up, which requires about two hours. At starting the speed may be 5 per cent below normal, and will gradually increase as the motor warms up, until the normal speed is attained. During this period the pump does not deliver its rated capacity, and if the service is intermittent, the pump may never deliver rated capacity as motor may not operate long enough at a time to reach full speed. This operating condition is not encountered with alternating-current drive.

76. It will be found in nearly all cases that the **starting duty of centrifugal pumps** will permit the use of practically any type of motor. However, there are features such as permissible starting current, frequency of starting, motor speed and speed variations which are important factors in selecting a motor. The **power-driven positive-displacement pump** with tightly packed stuffing boxes and pistons and full discharge head may require a **starting** torque equal to 125 to 250 per cent of the normal full-load torque, depending largely upon the care used in packing the pump. These starting requirements may be improved by the use of a by-pass which circulates the liquid from the discharge back into the suction.

77. General Motor Recommendations, Constant-Speed Service, Centrifugal Pumps:

Power Supply	Type of Motor
Direct current	Compound-wound
A.C. single-phase	Commutator-type
A.C. two- or three-phase	Squirrel-cage up to 500-hp. Slip-ring, 550-hp. and above 75-hp. and larger synchronous motors

78. For Positive Displacement Pumps:

Direct current	Compound-wound
A.C. single-phase	Commutator-type
A.C. two- or three-phase	5-hp. and smaller squirrel-cage 40-hp. and smaller wound-rotor, self-starting All capacities slip-ring motors

79. For Variable-Speed Service, Centrifugal Pumps and Positive-Displacement Pumps:

Direct current	Compound-wound
A.C. single-phase	Brush-shifting motor
A.C. two- or three-phase	Brush-shifting motor Slip-ring when speed reducing is small

80. The efficient operation of a motor-driven pump depends to a large extent upon the manner in which the **motor is controlled.** Data for specific recommendations for the control of motor-driven pumping equipment properly include the characteristics of the pump and motor and the conditions under which the pumping is to be done. Control equipment is divided into two classes, hand operated and magnetically operated, each of which may be subdivided into equipment for starting and equipment for starting and speed regulating.

81. Hand-operated control for starting or speed regulating should be used only when an operator is available.

82. Magnetic or **remote control** may be used with motors of any size. Magnetic equipments are available for operation from push-button stations when it is desired to start or stop the pump from points remote from the starter. Push-button control is suitable for all large and small motors.

83. Automatic magnetic equipments are available for operation by means of float switches, pressure switches, thermostats or pressure regulators. A float switch, pressure governor or thermostat is used for maintaining prescribed limits of liquid levels, pressures or temperatures. **Pressure-regulator controls** are for pumps that run for comparatively long periods and the speeds of which must be changed to conform to rapidly fluctuating demands for the liquid delivered by the pump.

84. In all cases it is best to consult the manufacturer of starting equipment as to the type and kind of starter to use. This information will be cheerfully given and the purchaser is always assured of having the most suitable equipment for his requirements.

85. Do not **select a motor** that will be overloaded in continuous service. Insulation deteriorates rapidly under excessive temperature. A motor will safely withstand a heavy overload for short

periods, but will burn out in continuous service under the same overload conditions.

86. For centrifugal pump drives, be sure that motor will come up to speed rapidly and will develop sufficient torque to start the pump effectively. (See Par. 59.) Otherwise the pump will lose its suction.

87. For small centrifugal pumps, the squirrel-cage motor is most commonly used. Up to 5 hp., they may be thrown directly across the line.

88. For mine service or for service where excessive moisture is encountered, the squirrel-cage induction motor is preferable to the slip-ring or brush commutator type. It is best, however, to consult the manufacturer in all cases where excessive moisture is met.

89. For driving positive displacement pumps the compound wound direct-current motor is recommended, and for alternating-current the wound-rotor or slip-ring motor.

90. Units of Electrical Measurements.—In practical work, the resistance and electromotive forces commonly met with are usually so large that if the absolute electromagnetic c.g.s. units were used, the resulting numerical values would be inconvenient. On the other hand, capacities are generally so small that their numerical values in c.g.s. units would be very small fractions. For practical use, therefore, certain multiples of the c.g.s. units have been adopted by international agreement.

91. When a difference in electrical potential exists between two points, there is said to exist an **electromotive force** (e.m.f.) or tendency to cause a current from one point to the other. This e.m.f. is analogous to the pressure caused by a difference in the level of two bodies of water connected by a pipe. The pressure tends to force the water through the pipe, and the e.m.f. tends to cause electricity to flow. The pressure acting on the water is measured in pounds per square inch; the e.m.f. acting on the electric current is measured in volts and is defined thus: A **volt** is that electromotive force which will maintain a current of one ampere in a circuit the resistance of which is one ohm.

92. The **ampere** is the practical unit denoting the strength of an electric current, or the rate of flow of electricity. It is analogous to the flow of water through a pipe measured in gallons per second. The ampere expresses the rate of transmission of electricity per

second, and is defined as: That strength of current or rate of flow which would be maintained in a circuit, the resistance of which is one ohm, by an electromotive force of one volt.

93. The electrical measure of quantity is the **coulomb,** and is the quantity of electricity conveyed at a constant current or flow of one ampere for one second. This unit is not so commonly used as the **ampere-hour.** The ampere-hour is the quantity of electricity conveyed at a constant flow of one ampere for one hour.

94. The practical unit of electrical resistance is the **ohm,** which is analogous to the resistance to the flow of water caused by the friction of the water against the inside of a pipe. The ohm is that resistance which will limit the flow of electricity under an electromotive force of one volt to one ampere.

95. In 1827, Dr. Ohm of Berlin discovered in the course of his experiments that when an electromotive force produces a current in any given circuit, the strength of the current is always proportional to the value of the electromotive force. This has since been known as **"Ohm's Law."** Expressed in terms of practical units it is: The number of amperes between two points of a circuit is equal to the number of volts of potential difference divided by the number of ohms of resistance between the same points, or

$$\text{Amperes} = \frac{\text{volts}}{\text{ohms}}$$

$$\text{Volts} = \text{amperes} \times \text{ohms}$$

$$\text{Ohms} = \frac{\text{volts}}{\text{amperes}}$$

96. The **watt** is the electrical unit of power, and is the power of a current of one ampere under a pressure of one volt. This unit of power is equal to one joule per second and is approximately 1/746 of a horsepower. The kilowatt (kw.) is equal to 1000 watts.

97. The **joule** is the electrical unit of work and is the energy expended in one second by one ampere in a circuit having a resistance of one ohm.

98. The **watt-hour** is the energy expended in one hour when the activity or power is one watt (3600 joules). The **kilowatt-hour** is the energy expended in one hour when the activity is one kilowatt, and is the product of the kilowatts times the hours. The

kilowatt-hour is the commercial unit of work and is equal to 1.34 hp-hr. or 2,654,280 ft-lb.

99. **Power-factor** is a term that is in daily use in connection with alternating-current motors, to indicate what fraction of the apparent watts, or volt-amperes, is effective power. It is expressed:

$$\text{Power-factor} = \frac{\text{effective power}}{\text{apparent power}} = \frac{\text{watts}}{\text{volts} \times \text{amperes}}$$

100. The **apparent power** is the product of the effective volts times the effective amperes. The term "cos Φ" is called the power-factor.

101. To **find the brake-horsepower** (P) of a direct-current motor:

$$P = \frac{\text{volts} \times \text{amperes} \times \text{efficiency of motor}}{746}$$

102. To find the brake-horsepower of an alternating-current motor:

$$P = \frac{\text{volts} \times \text{amperes} \times \cos \Phi \times \sqrt{N} \times \mu}{746}$$

Where μ is the motor efficiency; N the number of phases, and $\cos \Phi$ the power-factor of the motor.

103. A **direct current** is a flow in one direction only; that is, the current never reverses. It may change in value or it may pulsate, but it does not change its direction.

104. An **alternating current** is a flow that changes from a positive value to a negative value with a constant regularity. The complete set of values, one positive and one negative, through which the current passes, is called a cycle. The number of cycles per second is called the frequency. The standard frequencies in the United States are 25 cycles and 60 cycles. Alternating current is represented graphically in Fig. 7.

105. As a moving conductor approaches a pole of the field magnet, the e.m.f. rises in value as the conductor enters the strong field under the pole and falls in value as the conductor passes from under the pole. As the conductor passes a point between a north pole and a south pole the e.m.f. falls to zero. The duration of one cycle is shown by the figure, and this cycle repeats itself as the conductors pass by successive pairs of field poles.

106. A **circular mil** is a unit of area used in measuring the cross

section of wires, and is equal to $\frac{0.7854}{10^6}$ sq. in. The sectional area of a wire expressed in circular mils is equal to the square of its diameter in mils.

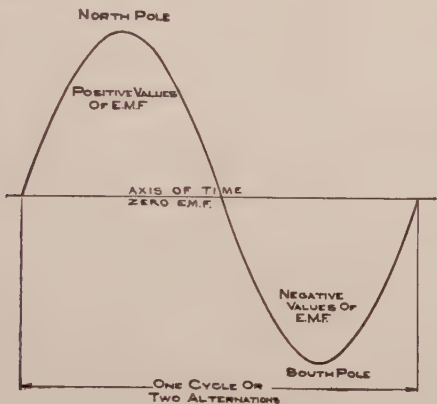


Fig. 7. Sine wave of electromotive force.

107. Electrical Equivalents Condensed

ONE WATT =	A rate of doing work 1.0 ampere at one volt 0.737 ft-lb. per sec. 44.238 ft-lb. per min. 2654.28 ft-lb. per hr. 0.00134 horsepower $\frac{1}{746}$ horsepower
ONE KILO-WATT =	A rate of doing work 737.3 ft-lb. per sec. 44238. ft-lb. per min. 1.34 horsepower
ONE WATT-HOUR =	A quantity of work 2654.28 ft-lb. 1.0 ampere-hour $\times$ one volt 0.00134 hp-hr.
ONE AMPERE-HOUR =	A quantity of current One ampere for 1 hour irrespective of voltage Watt-hour $\div$ volts

TORQUE = { Force moving in a circle
A force of one pound at a radius of one foot

$$\text{KILOVOLT-AMPERE (a.-c. unit)} = \frac{\text{Kilowatts}}{\text{Power-Factor}}$$

108. FULL-LOAD SPEEDS FOR ALTERNATING-CURRENT MOTORS

Based on 4-per cent slip

Number of Poles	Cycles								
	25	27	30	33⅓	40	42	50	60	100
2	1440	1560	1730	1920	2300	2420	2880	....	....
4	720	780	865	960	1150	1210	1440	1725	2880
6	480	520	575	640	770	807	960	1150	1920
8	360	390	433	480	575	605	720	862	1440
10	290	310	345	385	460	485	575	690	1150
12	240	260	288	320	385	403	480	575	960
14	205	222	247	275	330	346	412	492	822
16	180	195	216	240	287	302	360	431	720
18	160	171	192	214	256	268	320	384	640
20	142	156	173	192	230	242	287	345	575

109. AMPERE RATINGS OF A.C. AND D.C. MOTORS AT FULL LOAD

The current values given in this table are the average for motors of different makes, types, frequencies and speeds. A variation of 10 per cent above or below the values given may be expected.

HP.	D. C. Motors			A. C. Motors							
				Single Phase		*Two-Phase			Three-Phase		
	110 Volt	220 Volt	550 Volt	110 Volt	220 Volt	220 Volt	440 Volt	550 Volt	220 Volt	440 Volt	550 Volt
¼	2.4	1.2	0.5	5.	2.5	0.9	0.4	0.3	1.	0.5	0.4
½	4.5	2.3	0.9	6.8	3.4	1.6	0.8	0.6	1.8	0.9	0.7
¾	6.8	3.4	1.3	10.	5.	2.3	1.2	0.9	2.7	1.3	1.
1	8.8	4.4	1.8	13.3	6.6	3.	1.5	1.1	3.5	1.8	1.3
2	16.8	8.4	3.4	25.	12.4	5.6	2.8	2.2	6.5	3.3	2.6
3	24.5	12.3	4.9	36.	18.	7.4	3.7	3.3	9.5	4.8	3.8
5	40.	20.	8.	58.	29.	14.3	7.2	5.4	15.4	7.7	6.2
7½	59.	29.	11.8	85.	43.	19.4	9.7	7.8	22.4	11.2	9.
10	78.	39.	15.6	110.	55.	25.	12.5	10.8	29.	14.5	11.8
15	116.	58.	23.	162.	81.	37.	18.4	15.	42.	21.3	17.4
20	156.	78.	31.	208.	104.	48.	23.8	19.5	55.	28.	22.5
25	192.	96.	38.	258.	129.	59.	30.	23.	68.	34.	27.
30	230.	115.	46.	304.	152.	69.	35.	28.	80.	40.	32.
35	267.	133.	53.	356.	178.	81.	41.	32.	94.	47.	37.
40	304.	152.	61.	400.	200.	91.	46.	36.	105.	53.	42.
50	378.	185.	76.	492.	246.	112.	56.	45.	130.	65.	52.
60	452.	226.	90.	.....	.....	134.	67.	54.	155.	78.	62.
75	562.	281.	112.	.....	.....	166.	83.	67.	192	96.	77.
85	636.	318.	127.	.....	.....	185.	93.	75.	214.	107.	86.
100	748.	374.	150.	.....	.....	218.	109.	88.	252.	126.	101.
125	930.	465.	186.	.....	.....	268.	134.	108.	310.	155.	125.
150	1122.	561.	223.	.....	.....	320.	160.	128.	368.	184.	148.
175	1300.	650.	260.	.....	.....	367.	184.	149.	425.	213.	172.
200	......	742.	297.	.....	.....	418.	209.	169.	484.	242.	195.
225	......	835.	335.	.....	.....	467.	234.	189.	540.	270.	215.
250	......	925.	370.	.....	.....	515.	258.	207.	595.	288.	240.

*Ampere ratings are for 2-phase, 4-wire connection. In the case of 2-phase, 3-wire connection the current in the common leg is 1.4142 times the value given.

112. APPROXIMATE AMPERES PER TERMINAL FOR INDUCTION MOTORS

For Determining Size of Wires, Capacity of Fuses and Setting of Circuit Breakers

Hp.	110 Volts		220 Volts		440 Volts		550 Volts	1100 Volts	2200 Volts
	2 Ph.	3 Ph.	2 Ph.	3 Ph.	2 Ph.	3 Ph.	3 Ph.	3 Ph.	3 Ph.
½	3.3	3.7	1.7	1.8	.9	1.	.....	...	...
1	6.	6.5	3.	3.2	1.5	1.6	.....	...	..
2	10.5	12.	5.	6.	2.6	3.	2.5	...	...
3	15.	17.	7.5	9.	3.8	4.5	3.5	...	...
5	27.	30.	13.	15.	6.5	7.5	6.	...	...
7½	....	....	20.	22.	10.	11.	9.	...	...
10	....	....	25.	29.	12.5	14.	11.	...	...
15	....	....	35.	41.	18.	20.	16.	...	...
20	....	....	48.	55.	24.	27.	22.	...	...
25	....	....	54.	62.	27.	31.	25.	...	...
30	....	....	70.	81.	35.	40.	32.	16	8
40	....	....	95.	109.	47.	54.	44.	21	11
50	....	....	110.	127.	55.	64.	52.	27	13
75	....	....	165.	192.	83.	96.	77.	39	20
100	....	....	215.	248.	108.	124.	100.	50	25
150	....	....	320.	366.	160.	183.	147.	80	40
200	....	....	410.	475.	205.	237.	192.	98	49
250	....	....	510.	590.	250.	290.	237.	125	62
300	....	....	600.	700.	300.	350.	285.	150	75

113. DIMENSIONS, WEIGHT, AND DISTANCE OF COPPER WIRE

B. & S. Gage	Diameter in Mils (d) 1 mil = .001 in.	Area	Weight and Length		Resistance at 75° F. Ohms per 1000 ft.	Current Amperes		B. & S. Gage
		Circular Mils (d^2)	Lb. per 1000 Ft.	Ft. per Lb.		Exposed	Concealed	
0000	460.000	211,600.0	639.33	1.56	.049	300	175	0000
000	409.640	167,805.0	507.01	1.97	.062	245	145	000
00	364.800	133,079.0	402.09	2.49	.078	215	120	00
0	324.950	105,592.0	319.04	3.13	.098	190	100	0
1	289.300	83,694.0	253.88	3.95	.124	160	95	1
2	257.630	66,373.0	200.54	4.99	.156	135	70	2
3	229.420	52,634.0	159.03	6.29	.197	115	60	3
4	204.310	41,742.0	126.12	7.93	.248	100	50	4
5	181.940	33,102.0	100.01	10.00	.313	90	45	5
6	162.020	26,250.0	79.32	12.61	.395	80	35	6
7	144.280	20,817.0	62.90	15.90	.498	67	30	7
8	128.490	16,509.0	49.88	20.05	.628	60	25	8
9	114.430	13,094.0	39.56	25.28	.792	....	....	9
10	101.890	10,381.0	31.37	31.88	.999	40	20	10
11	90.742	8,234.1	24.88	40.20	1.260	....	....	11
12	80.808	6,529.9	19.73	50.69	1.589	30	15	12
13	71.961	5,178.4	15.65	63.91	2.003	....	....	13
14	64.084	4,106.8	12.41	80.59	2.526	22	10	14
15	57.068	3,256.7	9.83	101.65	3.186	....	....	15
16	50.820	2,582.9	7.80	128.17	4.017	15	5	16
17	45.257	2,048.2	6.19	161.59	5.066	....	....	17
18	40.303	1,624.3	4.91	303.76	6.388	10	....	18
19	35.890	1,288.1	3.89	257.42	8.055	....	....	19
20	31.961	1,021.5	3.08	324.12	10.158	5	....	20

114. CARRYING CAPACITY OF CABLES

Area Circular Mils	Current Amperes		Area Circular Mils	Current Amperes	
	Exposed	Concealed		Exposed	Concealed
200,000	299	200	1,200,000	1147	715
300,000	405	272	1,300,000	1217	756
400,000	503	336	1,400,000	1287	796
500,000	595	393	1,500,000	1356	835
600,000	682	445	1,600,000	1423	873
700,000	765	494	1,700,000	1489	910
800,000	846	541	1,800,000	1554	946
900,000	924	586	1,900,000	1618	981
1,000,000	1000	630	2,000,000	1681	1015
1,100,000	1075	673	....	.....	....

SECTION V

Direct-Acting Steam Pumps

(Figures refer to paragraph numbers)

SECTION V

Direct-Acting Steam Pumps

1. No machine has ever been produced for any purpose that can equal the direct-acting steam pump in the essential qualities of flexibility, simplicity, reliability and low cost.

2. **Types.**—Direct-acting steam pumps are of the positive displacement type with both steam and liquid pistons mounted on the same rod and operating together independently of any crank movement. They have been developed by experience into many subdivisions of type and pattern, each adapted to best meet some one set of service conditions. The best known and most commonly used types are the **"SIMPLEX"** (Reg. U.S. Pat. Off.) and the "Duplex," each subdivided into piston or plunger type as to liquid end.

3. A **simplex** or single pump as it is sometimes called, has one steam piston and one liquid piston or plunger, both double-acting.

4. A **duplex** pump is one having two steam pistons and two liquid pistons or their equivalent plungers, all of which are double-acting. A duplex pump is therefore equal to two simplex pumps when their liquid ends are of equal diameter and their piston speeds the same.

5. The duplex direct-acting steam pump was **invented by Henry R. Worthington** in 1859. It was a positive advance in the art of moving water under pressure and really began a new era in the practise of handling water. It permits of remarkable simplicity of construction, insures smoothness of working, efficiency of action and reliability for extended use whatever the pressure, the length of water column or the size of the apparatus employed.

6. The leading **characteristic** of this invention, in its simplest form, is that it is a direct-acting, non-rotative pump. Practically, the system consists of two independent pumps lying side by side, the motion of one pump actuating the steam valve of the other. The action produces an almost absolutely uniform delivery of water from the pumps.

7. Fig. 1 represents approximately the flow from a Worthington Pump at each point of the stroke. As soon as one pump begins to slow down at the end of its stroke, the other pump starts. By combining the flow it will be seen how uniform it is.

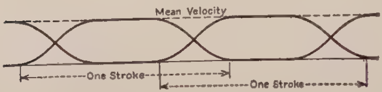


Fig. 1. Flow diagram of Worthington Duplex Pump.

8. A direct-acting steam pump is made up of two elements: (1) The steam end which supplies the motive power to (2) the liquid end which handles the water or other liquid that is to be pumped. The essential points of each will be briefly discussed under their separate headings.

9. The prominent and distinguishing feature of the Worthington Duplex **steam end** is its valve motion. Two steam cylinders are placed side by side, and so combined that one piston acts to give steam to the other, after which it completes its own stroke, and waits for its valve to be acted upon by the other piston before it makes the return stroke. This pause allows the liquid valves to seat quietly and produces a smooth-running pump. As one or the other of the steam valves is always open there is no "dead center" and, therefore, the pump is always ready to start when steam is admitted to the steam chest.

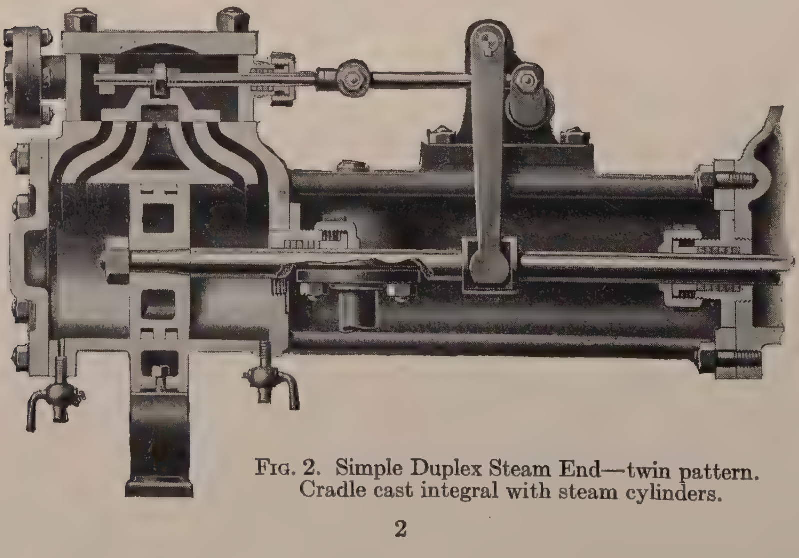


Fig. 2. Simple Duplex Steam End—twin pattern. Cradle cast integral with steam cylinders.

10. The duplex **valve motion** can be applied to simple high-pressure, compound or triple-expansion steam ends with slide, piston or semi-rotative steam valves. Figs. 2 and 3 illustrate simple high-pressure duplex steam ends. Motion is transmitted from each steam piston to the slide valve by a rockshaft mounted in the cross-stand. To one end of each rockshaft is fitted a lever, the lower end of which fits into or is attached to one of the cross heads on the corresponding piston rod. To the opposite end of the rockshaft is fitted a crank which is attached to the corresponding valve rod by means of a fork end and link. This combination of rockshafts, levers and cranks constitutes the operating mechanism of the steam valves.

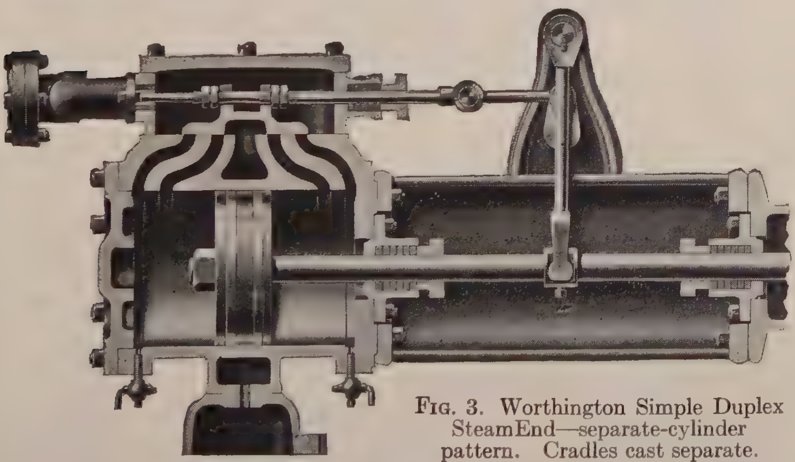


Fig. 3. Worthington Simple Duplex Steam End—separate-cylinder pattern. Cradles cast separate.

11. The most used steam end is the **simple duplex,** having one pair of double-ported high-pressure steam cylinders. This type of steam end is low in first cost and is always used where low operating costs are of little importance. Fig. 2 illustrates the twin type, with cradles cast integral with the steam cylinders. The twin construction makes possible the boring of both the right and left-hand cylinders at the same time, which guarantees the correct alignment. Where size and weight prohibit the twin construction, the cylinders and cradles are cast separate from each other, as shown by Fig. 3. This type of steam end is slightly more expensive than the twin type.

12. For installations where economy is of greater importance than first cost, the **compound steam end,** having one pair of high-pressure and one pair of low-pressure cylinders, is used. With a

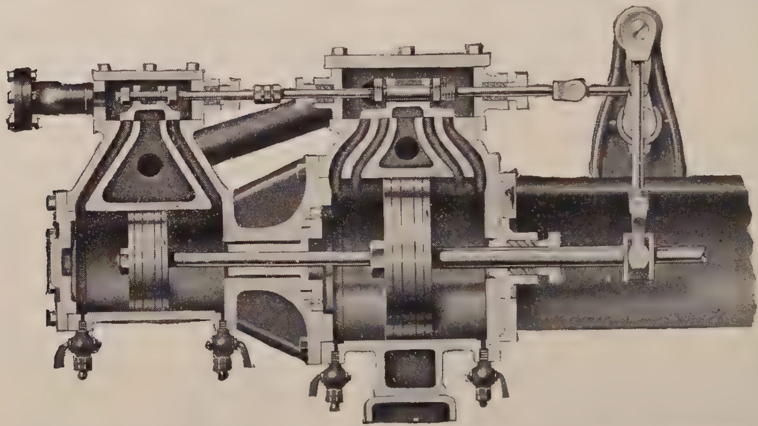


Fig. 4. Worthington Compound Duplex Steam End—12-in. stroke and under.

compound steam end, the saving over the simple duplex steam end is approximately 30 per cent when operating non-condensing and 50 per cent when operating condensing.

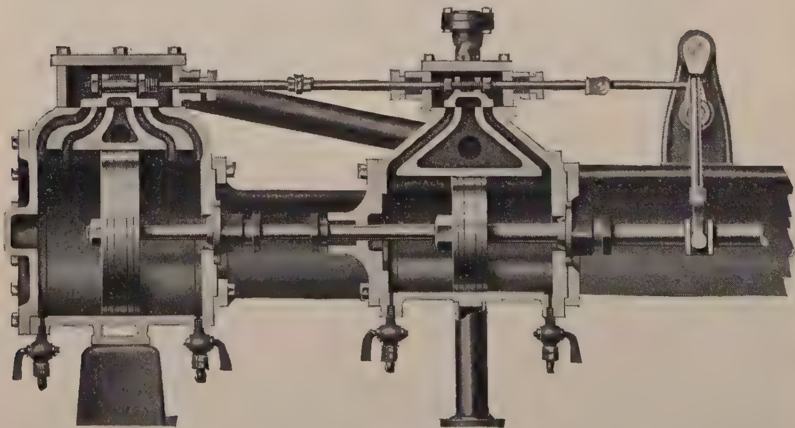


Fig. 5. Worthington Compound Duplex Steam End—15-in. stroke and over.

13. Fig. 4 shows the construction of Worthington compound steam ends of 12-in. stroke and under. The high-pressure cylinder is outboard and the low-pressure cylinder inboard. This form of

construction decreases the length overall, and eliminates four piston-rod stuffing boxes between the high- and low-pressure steam cylinders. This is a very compact, inexpensive type of compound steam end.

14. Compound steam ends of the **larger sizes** have the high-pressure cylinders inboard and the low-pressure cylinders outboard as shown by Fig. 5. On account of their weight, it is preferable to have the larger sizes of high-pressure cylinders on individual supports, thus insuring more perfect alignment than could be obtained with the high-pressure cylinders overhanging, as in the smaller sizes. This design also permits of the removal of all pistons without breaking pipe joints.

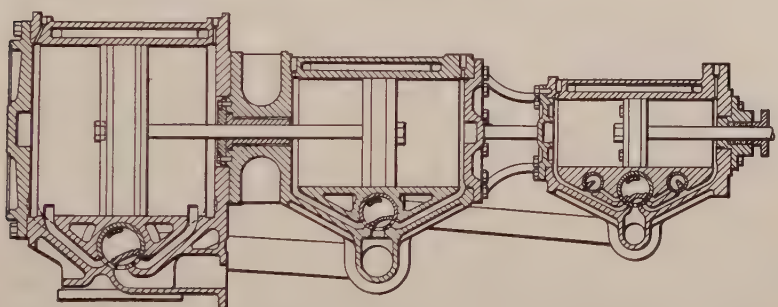


FIG. 6. Worthington Triple-expansion Steam End with semi-rotative steam valves.

15. Compound non-condensing steam ends have high-pressure cylinders with single **ports** and low-pressure cylinders with double ports. Compound condensing steam ends sometimes have double ports in both the high- and low-pressure cylinders when extra cushioning effect is desired.

16. Where economy is of the utmost importance the **triple-expansion steam end** with semi-rotative steam valves, as shown by Fig. 6, is used. The high-pressure cylinders are fitted with cut-off valves permitting adjustment to take care of variations in load. The steam cylinders are steam-jacketed and every provision is made for the most economical operation. The construction is such that each steam piston can be removed from the cylinder by simply removing the cylinder head. Pumps having triple-expansion steam ends are usually operated condensing.

17. Cushion Ports.—In order to **prevent** the **steam pistons from striking** the cylinder heads when the pump is operating at full speed, the double-ported or the five-ported cylinder, as it is sometimes called, has been adopted for simple steam ends, and for the low-pressure cylinder of compound steam ends. In this type of cylinder, the outside ports are for the admission of steam and the inside ports for the exhaust of the steam. The steam piston as it approaches the head covers the exhaust port and the remainder of the steam is trapped in the space between the piston and the head, where it compresses and forms a cushion preventing the piston from striking the head.

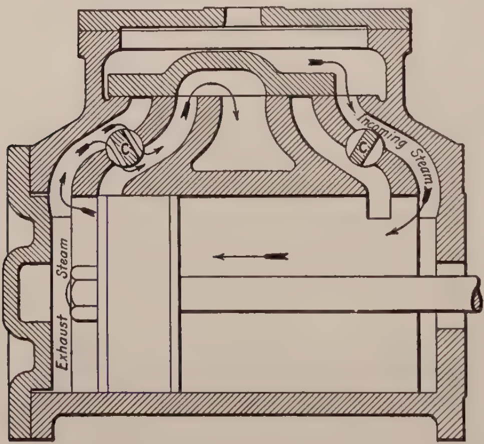


Fig. 7-8. Worthington Steam Cylinder with cushion valves.

18. Steam cylinders of 12 in. diameter and over are fitted with **cushion valves** in each end as shown by Fig. 8. These valves provide an adjustable steam cushion for the steam piston so as to insure a full stroke, and at the same time prevent the piston from striking the cylinder head. The cushion valves are regulated from outside the steam cylinder by means of a handwheel attached to the valve stem.

19. Compound condensing steam ends sometimes have cushion valves in both the high- and low-pressure cylinders. Compound non-condensing steam ends have cushion valves in the low-pressure cylinders only.

20. Triple-expansion steam ends may have cushion valves in either the intermediate or low-pressure cylinders, usually the low-pressure cylinders.

21. To better understand the **function of the cushion valve** we will suppose the steam piston to be traveling in the direction indicated by the arrow in Fig. 8. When the steam piston covers the opening

of the exhaust port, the volume of steam remaining in the cylinder cannot escape if the cushion valve is closed, but immediately forms a cushion. If the cushion valve is open, as shown, the steam flows by the admission port, through the cushion valve to the exhaust port, and the piston completes its stroke.

22. The amount of steam **cushioning** can be **regulated** by the simple adjustment of these valves. Perfect control of the stroke can be maintained under all service conditions. To lengthen the stroke open the cushion valve; to shorten the stroke close the cushion valve.

23. If the pump is operating at slow speed or under a heavy load, the cushion valves should be as wide open as possible without allowing the piston to strike the heads. If the pump is operating at high speed, or under a light load, the cushion valve should be opened only enough to prevent the pistons from striking the heads.

24. Steam Valves.—The steam valves of a duplex pump are without lap or lead. By giving no lap the possibility of the valve stopping in a position where all ports would be covered is eliminated.

25. The most used and the simplest of steam valves is the common **D-slide valve.** This type of valve is shown by Fig. 9 and is used for Worthington Steam Cylinders up to 25 in. diameter and for steam pressures up to 150 lb. gage. Slide valves may be used on cylinders up to and including 10 in. diameter for a total steam temperature of 450 deg. F. and above 10 in. for a total steam temperature of 400 deg. F. including superheat. It is difficult to lubricate the slide valve properly above these temperatures. Temperatures above 400-450 deg. F. warp the slide valve and cause excessive leakage of steam.

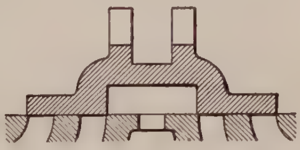


FIG. 9. Slide Steam Valve—D Type.

26. For steam at temperatures above the limits of the slide valve the balanced **piston** type of **steam valve** is recommended. The construction of this valve is shown by Fig. 10.

27. The piston steam valve is circular and there is no trouble from warping or distortion. Steam or boiler pressure exists on both ends

of the valve, giving a balanced condition, and reducing friction to that due to the weight of the valve itself plus sufficient tension in the piston rings to keep the valve steam tight.

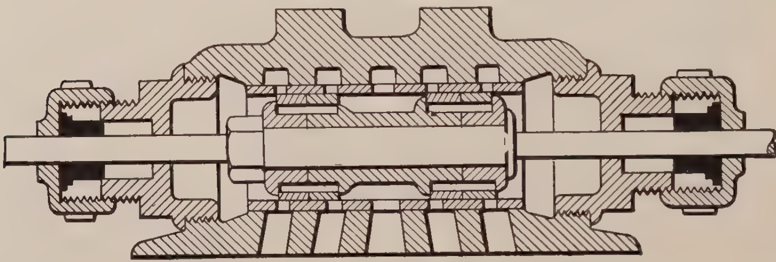


Fig. 10. Balanced Piston Steam Valve.

28. The **semi-rotative steam valve,** Fig. 11, is practically the same as the standard Corliss valve used on steam engines. This valve is usually located in the bottom of the steam cylinder and works in a cast-iron removable bushing in which the steam ports are accurately milled. The valve is operated by means of a tee-headed spindle fitted into a slot in the end of the valve. A valve bracket or bonnet with a stuffing box and gland is provided for the valve spindle. The rotary valve is always on its seat whether steam pressure is on or not. This type of valve is easily lubricated and can be used for moderately superheated steam.

Fig. 11. Semi-rotative Steam Valve.

29. In order to obtain smooth operation and to maintain full stroke with variations in capacity and pressure, a certain amount of **lost motion** between the steam piston and its valve is necessary. This lost motion allows the steam valve to pause at the end of its travel and give the cylinder it controls sufficient steam. A pump without lost motion will short stroke because the steam valve will begin to close before sufficient steam enters the cylinder to carry the piston to the end of its stroke.

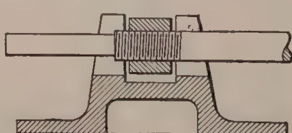


Fig. 12. Inside fixed lost motion.

30. Lost motion is caused in various ways. **Inside fixed lost motion** for small pumps is shown by Fig. 12. A valve-rod nut, the thickness

of which is less than its working space in the slide valve, is fitted in place by the pump manufacturer. This lost motion can only be changed by inserting a valve-rod nut of different thickness, which should never be necessary.

FIG. 13. Inside adjustable lost motion.

31. Fig. 13 illustrates inside **adjustable lost motion.** Here the lost motion can be changed by removing the steam-chest cover and adjusting the valve-rod nuts and lock nuts.

32. All pumps having inside lost motion are adjusted in the factory for the service conditions. It is unnecessary to make adjustments in the field unless the conditions of service are radically changed.

33. Steam ends having piston, or semi-rotative, steam valves are fitted with outside adjustable lost motion. The lost-motion link used with piston steam valves is shown by Fig. 14. The lost motion is obtained by the use of a slotted tube with adjusting nuts and lock nuts fitted over a link to which a tappet nut is riveted.

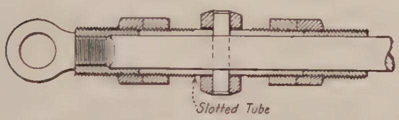


FIG. 14. Outside adjustable lost motion link.

34. Fig. 15 is a lost-motion link used with semi-rotative valve steam ends. The round-head set screw with its lock nut is adjusted to regulate the lost motion. This adjustment can be made without stopping the pump.

35. Cradles.—The cradles or connecting pieces between the steam ends and the liquid ends of Worthington Pumps are of semi-circular section combining strength, stiffness, and rigidity with minimum weight. The cross-stands or brackets for supporting the rockshafts are mounted on these cradles. With cast cradles of proper section, cylinders and rockshaft bearings are always in line.

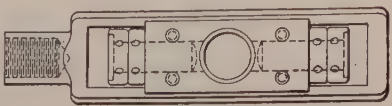


FIG. 15. Outside adjustable lost-motion link.

36. Tie Bars.—Rolled steel tie bars for connecting the steam ends and the liquid ends are not used by Worthington except in special cases where saving in weight is of importance. Tie bars are more flexible, and do not afford the same rigid connection between the steam and liquid cylinders as the cast cradle of proper section. Trouble is apt to be experienced in keeping the cylinders and the rockshaft bearings in proper alignment when tie bars are used.

37. Steam Pistons.—All Worthington **Steam Pistons** are fitted with self-adjusting packing rings. The piston bodies may be of the box type or of the body and follower type. The box type with snap rings is used for steam pistons up to 25-in. diameter. Larger-size pistons are of the body and follower type fitted with sectional rings consisting of overlapping segments set out by ribbon springs.

38. Liquid Ends.—The liquid ends in use today for positive-displacement pumps are of two general types or patterns, the inside-packed piston or plunger for low and moderate pressures, and the outside-packed plunger for high pressures.

39. These are further subdivided into the **submerged pattern,** with both the suction and discharge valves above the pistons or plungers,

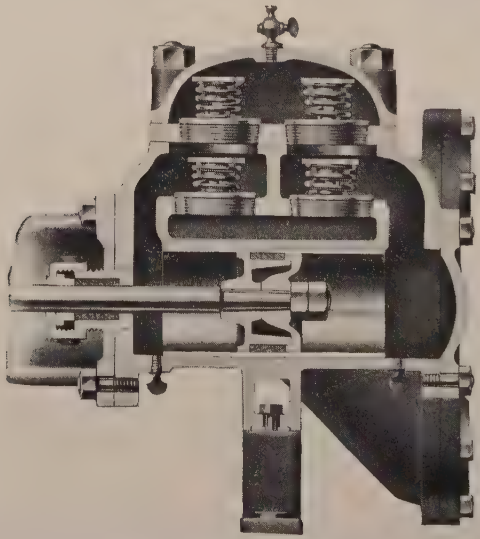


Fig. 16. Worthington Valve-plate Type Liquid End—submerged piston pattern. Driven bronze cylinder liners.

and the **straightway pattern,** with the suction valves below and the discharge valves above the pistons or plungers.

40. Piston-pattern liquid ends are built for discharge pressures up to 1000 lb. per sq. in.

41. Fig. 16 shows a Worthington **valve-plate** type, submerged-piston-pattern liquid end. The suction valve deck is cast integral with the cylinder. The discharge valve deck is separate so that it can be removed, and in this way access is gained to the suction valves. The liquid cylinder is fitted with driven bronze liners. The liquid pistons are fitted with fibrous packing. Fig. 17 illustrates the same type of liquid end fitted with removable liners held in place by binder rings. This construction is used for the smaller sizes of pumps.

42. The **turret-pattern** liquid end shown by Fig. 18 derives its name from the circular form of its valve chamber. This type is chiefly used in larger size pumps. Both the suction and discharge valve decks are cast integral with the cylinder. The suction valves are accessible through handholes in the sides of the valve

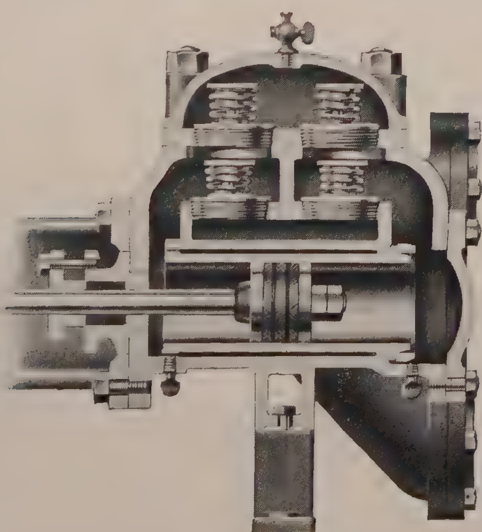


FIG. 17. Worthington Valve-plate Type Liquid End—submerged piston pattern. Removable cylinder liners.

chamber and the discharge valves by removing the cover plate on top of the valve chamber or through handhole plates bolted to the covers. The cylinders are fitted with driven bronze liners.

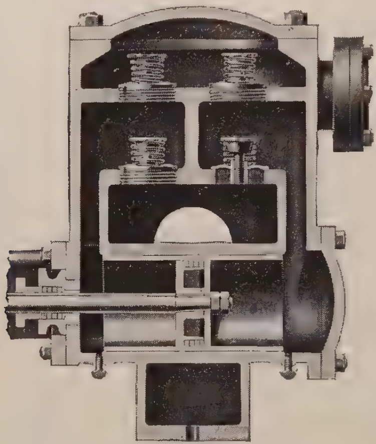


Fig. 18. Worthington Turret-type Liquid End—submerged piston pattern.

43. Fig. 19 shows the **straightway-type** liquid end with removable piston sleeves. With this design the liquid flows through the pump practically without change of direction. When the piston comes to a stop at the end of its stroke, the liquid, owing to its inertia, may continue to flow through the pump for a short time, which is a great advantage when pumping large quantities of liquid.

44. For heavy-pressure work the **packed-plunger liquid end** is used. For pressures up to 300 lb. per sq. in., the center-packed (See Fig. 20) or the end-packed type may be employed. For pressures above 300 lb. per sq. in., the end-packed type is used in practically all cases. On all packed-plunger pumps the stuffing boxes are visible and excessive leakage can be detected at once and stopped by tightening up on the stuffing-box glands while the pump is running.

47. For discharge pressures up to 2000 lb. per sq. in., the **pot-pattern** liquid end with packed plungers is principally used. This type is shown by Fig. 21 and Fig. 22. On account of the high pressures, the castings are made to as near a circular section as possible. The valve service is of the wing-guided type, easily accessible by removing the valve-pot cover.

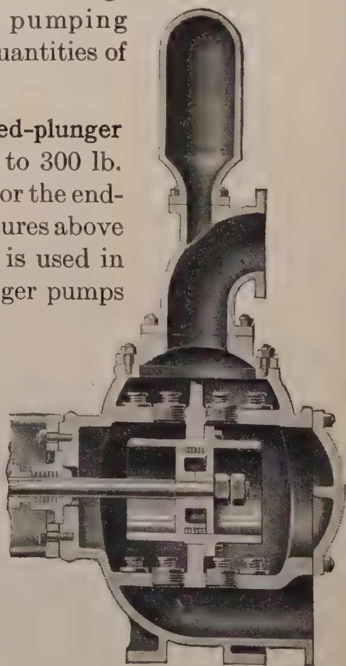


Fig. 19. Worthington Straightway Piston-pattern Liquid End. Removable cylinder liners.

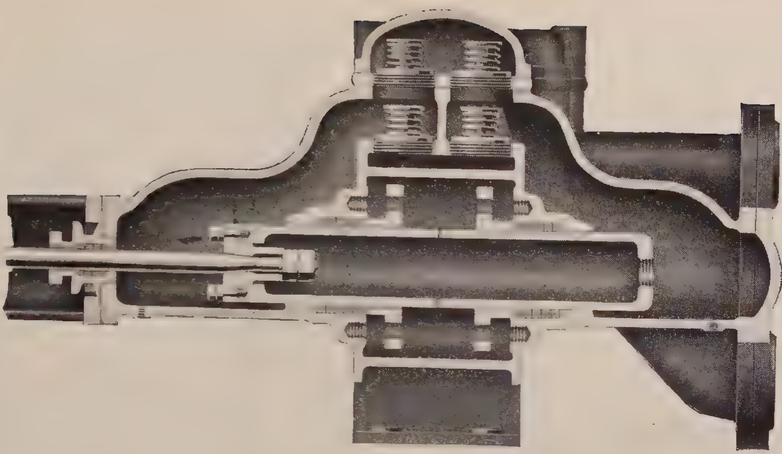


Fig. 20.

Center-Packed Plunger Liquid End. Valve-Plate Type.
Longitudinal Section.

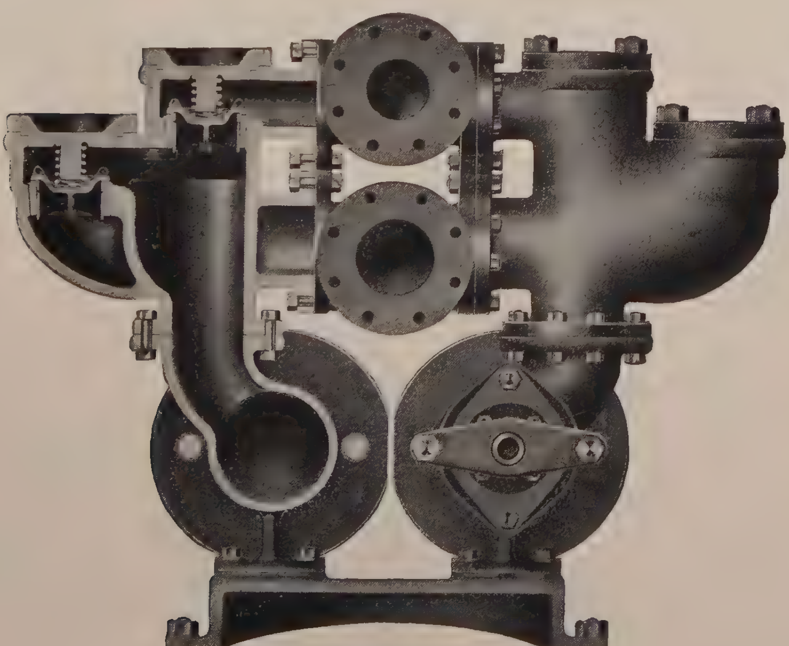


Fig. 21.

Pot-Pattern Liquid End. Cross-Section.

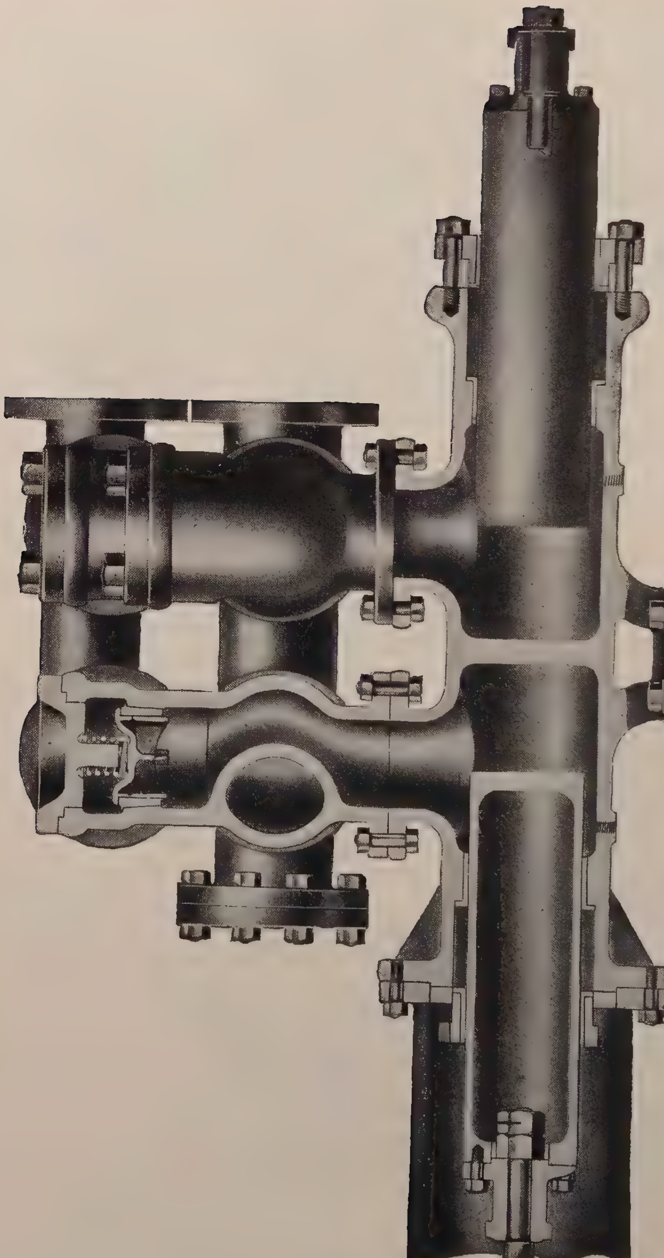


Fig. 22-24

Pot-pattern Liquid End with End-packed Plungers. Longitudinal Section

48. Liquid ends of the type shown by Fig. 86 are used in many cases for discharge pressures from 1000 to 2000 lb. per sq. in., but are designed especially for discharge pressures above 2000 lb. per sq. in. The cylinders are of **forged steel,** having the liquid passages and valve chambers drilled from the solid billet. The liquid valves are of the wing-guided type. Each valve is in a separate chamber, readily accessible by the removal of its individual valve-chamber plug. For heavy-pressure work, forged-steel liquid ends are superior to castings, not only on account of their greater strength, but on account of the dependability of the material and the practical elimination of all chances of leakage developing from porosity of the metal.

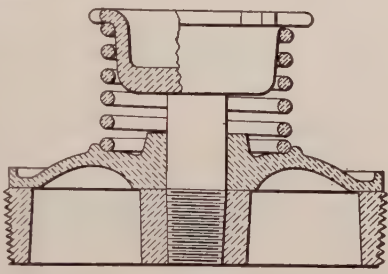


Fig. 25. Disk Type Metal Valve.

49. Liquid valves are roughly divided into three types, the disk valve for general service and thin liquids, the wing-guided valve for high pressures, and the ball valve for thick or viscous liquids.

50. The Worthington standard **Disk Valve** with one-piece cast-bronze guard and cylindrical coiled bronze wire spring is shown by Fig. 25. The valve spring is centered by the guard which also guides the valve and limits its lift. The seat is screwed into the deck on a taper thread of fine pitch. The valve is of bronze having a ground joint with its seat.

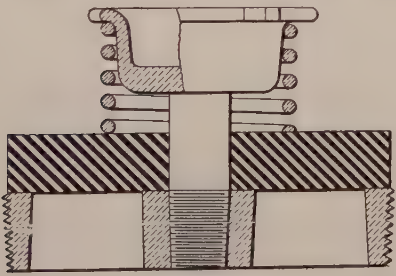


Fig. 26. Rubber Disk Valve.

51. Fig. 26 shows a valve with a plain **rubber disk.** For low service up to 75 lb. per sq. in., medium soft rubber valves are used. For pressures up to 150 lb. per sq. in., medium hard rubber or bronze valves may be used. For pressures above 150 lb. per sq. in., if rubber is used, it must be a hard rubber of special composition.

52. Fig. 27 shows a disk valve used in pumps for **elevator service.** A heavy bronze backing plate is used in conjunction with a rubber valve, giving in effect a rubber-faced bronze valve. The valve stem is of rolled bar bronze with a cast bronze guard held in position by a bronze lock nut.

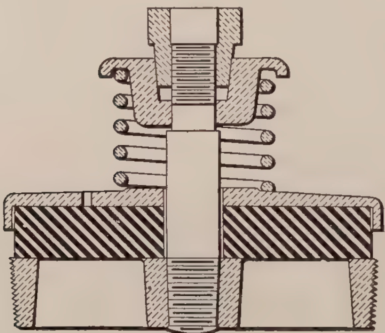


Fig. 27. Rubber Disk Valve, for elevator service.

53. Fire pumps built to the specifications of The Associated Factory Mutual Fire Insurance Companies are fitted with disk valves of the type shown by Fig. 28. The valve stem is of bar bronze fitted with a cast-bronze guard and lock nut. The valve disk is of rubber, protected from the cutting action of the spring by a special backing plate.

54. The seat is of bronze with a lug opposite each rib, these lugs being expanded out after the seat is inserted. The spring is guided by the groove in the valve guard and the bead on the valve plate.

55. The Worthington **"Seal" Valve** shown by Fig. 29 marks a distinct advancement in pump valve design. It is entirely different from any other pump valve and incorporates several new and **distinctive features,** the most prominent of which are:

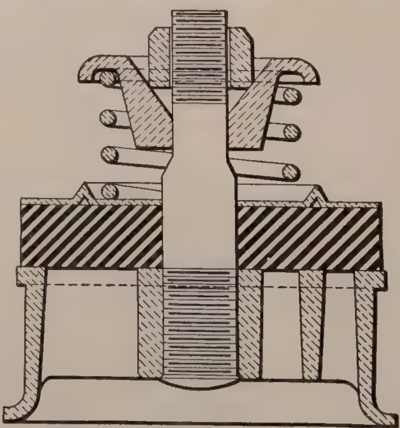


Fig. 28. Rubber Disk Valve for Underwriters' Fire Pump.

(1.) **Absolute tightness when closed.** When the valve is on its seat a flexible rubber seal affords a perfect seal at both the outer rim and the hub of the seat, preventing leakage. It, therefore, increases the efficiency of the pump, decreases the cost of pumpage and maintains pump capacity.

55. (2.) **Simplicity of construction.** Screws or bolts are not used in the construction of the Worthington Seal Valve for holding the rubber seal in position. There are no rubber rings to become loose from retainers; no screws or nuts to loosen up and cause trouble, and no bushings or rotating elements to wear on the stem.

(3.) **Light weight.** The moving parts of the Worthington Seal Valve are of light, rigid construction. The light weight assures a smooth, quiet running pump, free from the noisy pound or chattering of pump valves. The rigid construction prevents distortion and consequent leakage.

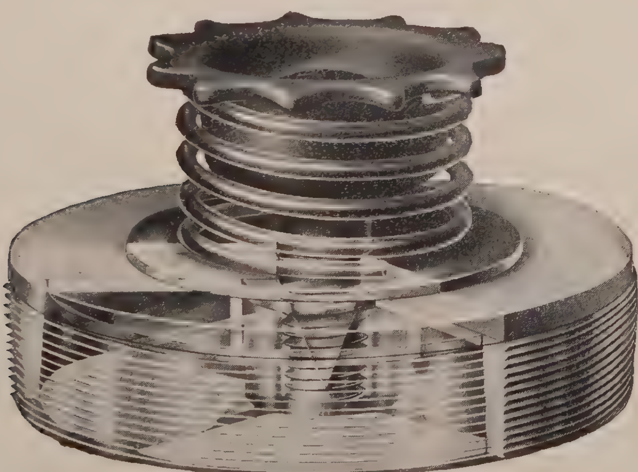


Fig. 29. Worthington Seal Valve.

(4.) **Long life.** The entire load on the Worthington Seal Valve is carried by the bottom plate, which is supported by the arms of the seat when the valve is closed. The top of the bottom plate forms a level seating surface for the seal, which prevents its warping or cracking and greatly prolongs the life of the valve. When examined after one year's service under 200 lb. per sq. in., seal valves were found to be tight and in excellent condition.

(5.) **Low cost and ease of replacements.** It is not often that **replacements** will be required for Worthington Seal Valves, but when such replacements become necessary, the

operation is a short, easy one. The valve stem is first removed, after which the spring, backing plate, seal or bottom plate can be repaired or renewed. The parts are then replaced and the stem screwed home, completing the operation. All parts are made to gage and new parts can be supplied from stock at small cost.

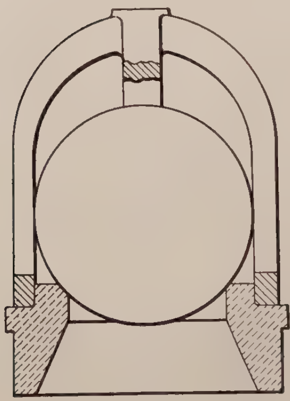


Fig. 30. Ball Valve for handling thick liquids.

57. Pumps for handling tar or similar thick liquids are fitted with **ball valves** of the type shown in Fig. 30. Suitable cages are provided for guiding the discharge valve and to limit its lift. Guides for the suction valve are cast on the bottom of the discharge-valve seat when necessary. The valves may be of rubber, bronze or iron as conditions require. With ball-valve service, the passage through the seat is free and unobstructed, due to the absence of cross ribs.

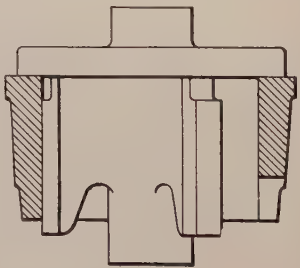


Fig. 31. Wing-guided Valve with flat face.

58. Pressure pumps of all types are usually fitted with **wing-guided valves.** This valve derives its name from the wings on the bottom of the valve which guide the valve in its seat. The wing valve may have a flat face as in Fig. 31 or a bevel face as in Fig. 32 or a leather or rubber face as in Fig. 33. This type of valve has a comparatively low lift, and is easily ground to a seat for all heavy pressures.

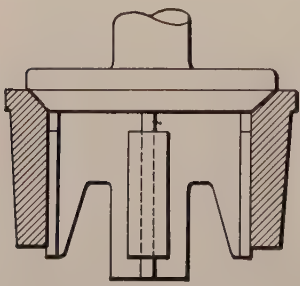


Fig. 32. Wing-guided Valve with bevel face.

Other types of valve service are described under "Simplex Pumps" and under "Oil Pumps."

59. Valve Area.—The area of the liquid valves is given in square inches and in

percentage, the percentage being the ratio of the valve area to the area of the liquid piston or plunger. The valve area is always given as the total of all the suction or discharge valves in one compartment of the liquid end. In a duplex pump there are four suction and four discharge compartments. There is generally the same number of valves in each compartment, and valve area may refer to the area of either the suction or the discharge valves.

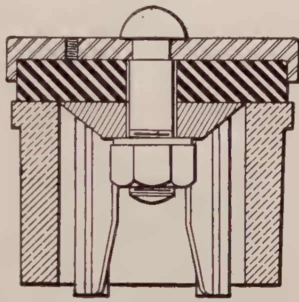


FIG. 33. Wing-guided Valve with leather or rubber face.

60. For water and **thin liquids** a valve area of 50 per cent for 100 ft. per min. piston speed gives a good smooth operating pump and a velocity of 200 ft. per min. through the valves. A maximum velocity of 222 ft. per min. through the valves is permissible for thin liquids, which would permit of a valve area of 45 per cent for 100 ft. per min. piston speed.

61. Thick, viscous liquids like tar, asphalt base crude oils, etc., require greater valve area than the thin liquids. The actual valve area will depend to a certain extent upon the viscosity of the liquid, and will range from 75 to 100 per cent of the piston area regardless of the piston speed, which is also controlled by the viscosity of the liquid.

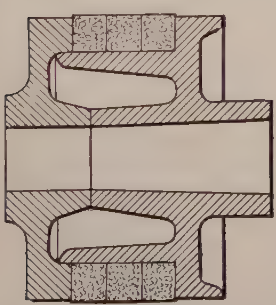


FIG. 34. Body and follower type of liquid piston, with fibrous packing.

62. Liquid Pistons.—Fig. 34 shows the body and follower type of liquid piston used in Worthington Duplex Piston-pattern Pumps. This type of piston can be repacked without removing it from the cylinder, by simply removing the follower. Solid pistons are very special and are designed for the particular conditions of service to which they are applied.

63. Many kinds and **types of packing** are available for liquid pistons. For low service (cold water) a good grade of flax is very satisfactory. For general service,

either hot or cold water, Worthington "Ankorite" packing is the best the market affords. Fig. 34 shows a liquid piston packed with "Ankorite" packing.

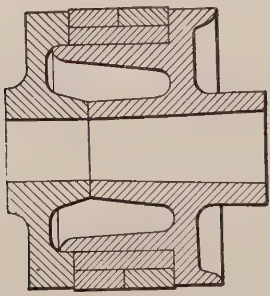


Fig. 35. Body and follower type of piston equipped with three-ring metal packing.

64. For hot or cold oil, tar, etc., the "three-ring" metallic packing is most generally used. This packing consists of an inside spring ring called the bull ring, and two outside split rings, as shown by Fig. 35. Three-ring packing can be made of either cast iron or bronze as conditions require. For cold distillates, gasoline, etc. the leather-cup packing, Fig. 36, is sometimes used. The cups may be of a good grade leather or rawhide. Rawhide is to be preferred for gasoline.

65. Plungers.—A plunger is a long barrel with closed ends, working through outside stuffing boxes, as shown by Fig. 37. Worthington plungers are always of solid metal and are made from hard cast iron or bronze, the material depending upon the liquid to be pumped. Cast-iron bronze-covered plungers are seldom satisfactory on cold-water service, as it is almost impossible to obtain the perfect fit necessary between the plunger and its cover. They are absolute failures on hot water, as the difference in the expansion of the two metals when subjected to high temperature will cause the bronze sleeve to come loose. All Worthington plungers are ground and polished.

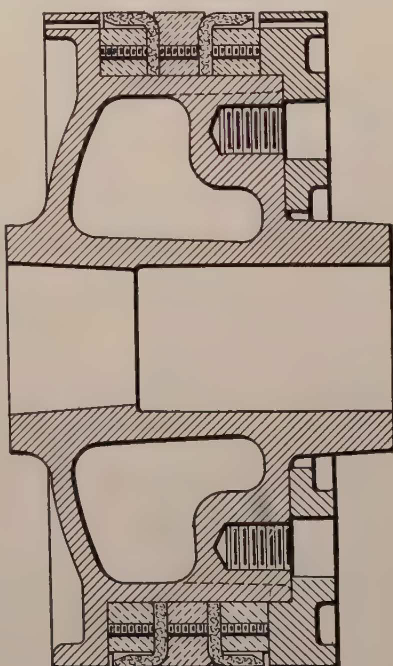


Fig. 36. Body and follower type of piston with leather-cup packing.

66. Selecting a Steam Pump.—Considerable care must be used in selecting the proper size and type of steam-driven pump to use for a given operating condition. The question of normal and maximum capacity in gal. per min., the nature of the liquid and its temperature, the suction lift or head, the discharge pressure in lb. per sq. in., the piston speed in ft. per min., and the number of hours per day the pump is to operate, must all be carefully considered.

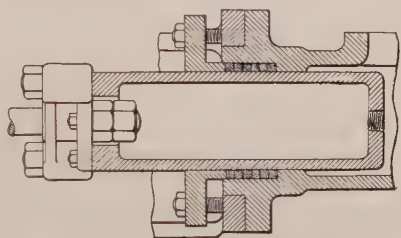


Fig. 37. Liquid Plunger with stuffing box packed from the outside.

67. To calculate the **size of a pump** always reduce the capacity to terms of U. S. gallons per minute (gal. per min.). The capacity of a pump is the actual gal. per min. delivered and is equal to the displacement minus the slip. For small-diameter pistons or plungers the area of the rod must be deducted when it extends into the liquid cylinder, as in packed-piston pumps and in center-packed plunger pumps. The area of tail rods must also be deducted if they are used. When plunger pumps have outside crossheads and rods no allowance is made for rods.

68. The **slip** in a pump properly packed is small, usually 3 to 5 per cent. For very accurate calculations it is, of course, necessary to know the diameter of the rods and to deduct this area from the piston or plunger area. The slip should be calculated after the pump is tested, and the actual quantity delivered measured by a carefully calibrated meter or weir. For commercial purposes this accuracy is unnecessary, as it has been found from many tests that 3 to 5 per cent allowance is ample to cover the loss by slip.

69. The **displacement** of a single, double-acting piston can be **calculated** from the formula

$$D = \frac{AP12}{231} = 0.0408\, d^2P$$

where D is the displacement in U. S. gallons per minute; A the area of piston or plunger in square inches; P the piston speed in feet per minute, and d the diameter of the piston or plunger in inches. For a simplex single-acting pump divide D by 2. For a duplex double-acting pump multiply D by 2.

70. The **diameter** of a single double-acting liquid **piston or plunger** to give a desired displacement may be found from the formula

$$d = 4.95\sqrt{\frac{D}{P}}$$

For a duplex pump divide D by 2.

71. To save time in making calculations for capacity, Table 74 gives the capacities of one double-acting piston or plunger of different diameters for a piston speed of 100 ft. per min. Capacities at other speeds may be easily calculated from this table.

72. The capacity of a pump having voluntary valves is limited by the number of times the valves will open and close smoothly and quietly, and not by the **piston speed** in ft. per min. Worthington pumps are rated at piston speeds, resulting from years of observation of pumps in actual operation under all conditions of service. These speeds, which are given in the following table, insure smooth, quiet operation of the liquid valves and minimum reversals of the pistons, both of which add to the durability and reliability of the pump under continuous operating conditions. To obtain the best results from any installation it is advisable in all cases to adhere as closely as possible to the **piston speeds recommended** by the pump manufacturer.

73. PISTON SPEEDS FOR DIRECT-ACTING PUMPS IN FEET PER MINUTE—AVERAGE CONDITIONS OF INSTALLATION

Stroke in Inches	General Service	Light Heads Low Service	Boiler-Feed	Pressure Pumps
3	28	25	18	20
4	33	30	22	24
5	38	34	24	26
6	40	36	26	29
8	47	43	30	33
10	58	52	38	42
12	60	54	39	43
15	75	68	49	54
18	90	81	59	65
24	100	90	65	72
36	120	108	78	86

NOTE: For emergency service these speeds may be increased 30 per cent.

74. DISPLACEMENT IN U. S. GALLONS OF ONE DOUBLE-ACTING PISTON OR PLUNGER COMPUTED FOR A PISTON SPEED OF 100 FT. PER MIN. NO ALLOWANCE MADE FOR SLIP

Diameter of Pump or Plunger in Inches	U. S. Gallons per Minute	U. S. Gallons per Hour	U. S. Gallons per 24 Hours	Diameter of Pump or Plunger in Inches	U. S. Gallons per Minute	U. S. Gallons per Hour	U. S. Gallons per 24 Hours
1	4.07	244.7	5875	7¾	245	14700	352300
1¼	6.37	382.5	9180	8	261	15667	376011
1½	9.18	550.8	13219	8¼	277	16660	399852
1¾	12.49	749	17992	8½	294	17688	424512
2	16.31	979	23500	8¾	312	18741	449978
2¼	20.6	1239	28180	9	330	19828	475887
2½	25.5	1530	36720	9¼	349	20944	502668
2¾	30.8	1851	44424	9½	368	22092	530208
3	36.7	2203	52878	9¾	388	23280	558720
3¼	43.1	2586	62064	10	408	24480	587518
3½	49.9	2998	71971	10¼	428	25716	617184
3¾	57.3	3442	82619	10½	449	26989	647789
4	65.2	3916	94002	10¾	471	28290	678960
4¼	73.7	4422	106128	11	493	29616	710784
4½	82.6	4957	118971	11¼	516	30986	743677
4¾	92	5523	132552	11½	539	32374	776993
5	102	6120	146880	11¾	564	33795	811080
5¼	112	6745	161934	12	587	35251	846046
5½	123	7404	177696	12¼	612	36735	881640
5¾	134	8093	194248	12½	637	38250	918000
6	146	8812	211511	12¾	663	39816	955584
6¼	159	9562	229500	13	689	41370	992880
6½	172	10344	248256	13¼	716	42972	1031328
6¾	185	11152	267660	13½	743	44610	1070640
7	200	11995	287884	13¾	771	46278	1110672
7¼	214	12867	308808	14	799	47980	1151536
7½	229	13769	330478	14¼	828	49704	1192896

NOTE: To compute the capacity of a duplex pump multiply the quantity given in the above table by 2. For a triplex (3) single-acting pump multiply the quantity given in table by 1½.

74. DISPLACEMENT IN U. S. GALLONS OF ONE DOUBLE-ACTING PISTON OR PLUNGER COMPUTED FOR A PISTON SPEED OF 100 FT. PER MIN. NO ALLOWANCE MADE FOR SLIP. (Concluded

Diameter of Pump or Plunger in Inches	U. S. Gallons per			Diameter of Pump or Plunger in Inches	U. S. Gallons per		
	Minute	Hour	24 Hours		Minute	Hour	24 Hours
14½	858	51648	1235232	21	1799	107952	2590848
14¾	887	53256	1278144	21¼	1842	110538	2652912
15	918	55070	1321915	21½	1886	113154	2715696
15¼	949	56928	1366272	21¾	1930	115800	2779200
15½	980	58800	1411200	22	1974	118482	2843568
15¾	1012	60720	1457280	22¼	2020	121194	2908656
16	1044	62668	1504046	22½	2065	123924	2974176
16¼	1077	64638	1551312	22¾	2111	126696	3040704
16½	1110	66642	1599408	23	2158	129492	3107808
16¾	1144	68676	1648224	23¼	2205	132324	3175776
17	1179	70752	1698048	23½	2253	135186	3244464
17¼	1214	72840	1748160	23¾	2301	138078	3313872
17½	1249	74964	1799136	24	2349	140958	3382992
17¾	1285	77124	1850976	24¼	2399	143952	3454848
18	1322	79314	1903550	24½	2449	146958	3526992
18¼	1359	81528	1956672	24¾	2499	149952	3598848
18½	1396	83778	2010672	25	2550	152994	3671856
18¾	1434	86060	2065449	25½	2653	159179	3820300
19	1473	88368	2120832	26	2758	165484	3971630
19¼	1511	90660	2175840	26½	2865	171908	4125800
19½	1552	93120	2234880	27	2974	178457	4282967
19¾	1590	95400	2289600	27½	3085	185130	4443125
20	1632	97920	2350080	28	3199	191922	4606125
20¼	1673	100380	2409120	28½	3314	198838	4772118
20½	1714	102840	2468160	29	3431	205876	4941028
20¾	1756	105396	2529504	30	3672	220320	5287675

NOTE: To compute the capacity of a duplex pump multiply the quantity given in the above table by 2. For a triplex (3) single-acting pump multiply the quantity given in table by 1½.

75. The maximum **discharge pressure** in pounds per square inch or the equivalent discharge head in feet for which various types of pumps have been designed are always listed by the manufacturer. The castings or forgings, bolting and flanges are amply strong for these working pressures. In actual service these maximum pressures should not be exceeded without first consulting the manufacturer.

76. In calculating the size of a steam end required for a given service, it is necessary to know the **plunger load** and the mechanical efficiency of the pump. The plunger load is equal to the area of the liquid piston or plunger in square inches multiplied by the total dynamic head* expressed in pounds per square inch.

77. Owing to the mechanical friction of the steam and liquid ends, the plunger load is only a part of the total load to be overcome by the steam. This total load, known as the **steam-piston force,** is:

$$\frac{\text{plunger load in pounds}}{\text{mechanical efficiency}}$$

78. Mechanical Efficiency.—The **mechanical efficiency** of a pump varies with the stroke and with the type of liquid end. A packed-plunger pump with four stuffing boxes will have a greater friction loss than a piston pump with two inside-packed pistons. The mechanical efficiency of any pump can only be determined by actual test. The accompanying table, par. 81, which is based on a number of tests of different types and sizes of pumps, will indicate the efficiency that should be obtained when the pump is in good operating condition.

79. When calculating the **size of a steam end,** we recommend that the mechanical efficiency be taken at 90 per cent of the value given in table, par. 81. This will take care of any drop in steam pressure, or any unforeseen conditions.

80. A direct-acting pump in its simplest form takes steam during the entire length of its stroke. For this reason all steam-piston forces are referred to the area of the high-pressure steam cylinder. In compound and triple-expansion steam ends, the high-pressure cylinder, in the absence of a cut-off, measures the amount of steam that is to pass through the steam end and determines the power of the steam end in a greater degree than any of the re-

*See Section I, par. 43.

maining cylinders. The size of the low-pressure cylinder, between certain limits, has very little influence on the steam-piston force, and its size is often determined by commercial considerations such as initial cost, existing patterns, etc.

81. MECHANICAL EFFICIENCIES IN PER CENT STEAM PUMPS OF VARIOUS TYPES

Stroke Inches	Piston Pattern to 250 lb. per sq. in.	Packed Plunger to 300 lb. per sq. in.	Pressure Pumps to 1000 lb. per sq. in.	Pressure Pumps above 1000 lb. per sq. in.	Thick Liquids
3	50	47	45	39	39
4	55	52	50	43	43
5	60	57	54	47	47
6	65	61	58	51	51
8	70	66	63	55	55
10	75	71	67	58	58
12	77	73	69	60	60
15	80	76	72	62	62
18	82	78	74	64	64
24	85	81	77	66	66
36	87	83	79	68	68

82. The **mean effective pressure** for different types of steam ends can be found from the following formula:

(1) Simple $\text{m.e.p.} = p - p_b$

(2) Compound $\text{m.e.p.} = 2\,p - \dfrac{p}{R} - p_b\,R$

(3) Triple expansion $\text{m.e.p.} = 3\,p - 2\dfrac{p}{R_2} - p_b\,R_1$

where: m.e.p. is the mean effective pressure referred to the high-pressure cylinder; p the absolute pressure in high-pressure cylinder (boiler pressure in pounds per square inch plus 10, allowing 4.7 for wire drawing and port losses); p_b the back pressure (absolute) = 16 lb. per sq. in. for non-condensing steam ends

= 6 lb. per sq. in. for compound condensing steam ends

= 5 lb. per sq. in. for triple condensing steam ends, and

R the ratio of cylinder areas. In a compound pump the ratio should not exceed 4, corresponding with 1/4 cut-off in a steam engine, which is considered the most economical point.

For triple-expansion steam ends.

$$R_1 = \frac{\text{Low-pressure area}}{\text{High-pressure area}} \text{ (should not exceed 8)}$$

$$R_2 = \frac{\text{Intermediate-pressure area}}{\text{High-pressure area}} \text{ (should not exceed 3)}$$

83. The approximate values of R; R_1; R_2 may be found from the formula

$$R = \sqrt{\frac{p}{p_b}} \quad \text{for compound steam ends}$$

$$R_1 = \sqrt[3]{\left(\frac{p}{p_b}\right)^2} \quad \text{for triple expansion steam ends}$$

$$R_2 = \sqrt{R_1}$$

84. These values should be used with discretion. Above certain limits the increased first cost of steam ends with large-diameter low-pressure cylinders more than offsets the gain in steam economy.

85. All **manufacturers** have developed commercial sizes of compound and triple-expansion steam ends with fixed ratios of cylinders. It is best in all cases to follow such **standards**, as they are proportioned for maximum steam economy with minimum initial cost, and it is these two factors combined that determine the commercial value of the steam end. A tabulation of standard Worthington compound and triple-expansion steam ends with steam-piston forces for various steam pressures will be found in par. 93, to par. 95.

86. To illustrate the **application of the formulas** given on the preceding pages we will consider a pump for 800 gal. per min.; discharge pressure 150 lb. per sq. in.; suction lift 16 ft.; steam pressure 110 lb. per sq. in. gage.

87. Assume 18-in. stroke **outside-packed plunger pump.** From table, par. 73, recommended piston speed is 90 ft. per min. Capacity = 800 gal. per min. plus 5 per cent slip = 840 gal. per min. Total dynamic head = 150 lb. per sq. in. plus 16 ft. suction (7 lb.) = 150 plus 7 = 157 lb. per sq. in.

Fig. 38.

The Large Stock of Worthington Pumps Means Prompt Shipment.

Plunger diameter formula

$$d = 4.95\sqrt{\frac{420}{90}} = 4.95 \times 2.16 = 10.69$$

Use 10¾-in. diameter liquid piston or plunger.

Plunger load = area plunger × t.d.h.
= 90.76 × 157
= 14,249 lb.

For the packed plunger 18-in. stroke, mechanical efficiency is 78 per cent (from table, par. 81).

78 per cent × 90 = 70.20, say 71 per cent

$$\text{Steam-piston force} = \frac{14{,}249}{0.71} = 20{,}069 \text{ lb.}$$

For simple pump
m.e.p. = (110 + 10) − 16 = 104 lb.
The area of the steam cylinder will be

$$\frac{20{,}069}{104} = 192.9 \text{ sq. in., or } 15\tfrac{5}{8}+, \text{ diameter in inches.}$$

The next largest commercial size is 16 in., so that the proper size for a duplex-packed plunger pump with simple steam cylinders is 16 by 10¾ by 18.

Compound non-condensing.

$$R = \sqrt{\frac{p}{p_b}} = \sqrt{\frac{120}{16}} = 2.74$$

$$\text{m.e.p.} = 2 \times 120 - \frac{120}{2.74} - 16 \times 2.74 = 152.7 \text{ lb. per sq. in.}$$

The steam-piston force is the same as before, viz.; 20,069 lb. and the area of the high-pressure cylinder will be

$$\frac{20{,}069}{152.7} = 131.4 \text{ sq. in., or } 12\tfrac{7}{8}+, \text{ diameter in inches.}$$

The area of the low-pressure cylinder will be 131.4 × 2.74 = 360 sq. in., or 21⅜ +, diameter in inches. The nearest commercial size steam end has 14-in. high-pressure and 22-in. low-pressure cylinders and the size of the compound non-condensing pump would be 14 and 22 by 10¾ by 18.

88. A compound condensing steam end would be **calculated** the same way except that the value of R would be taken as 4, which

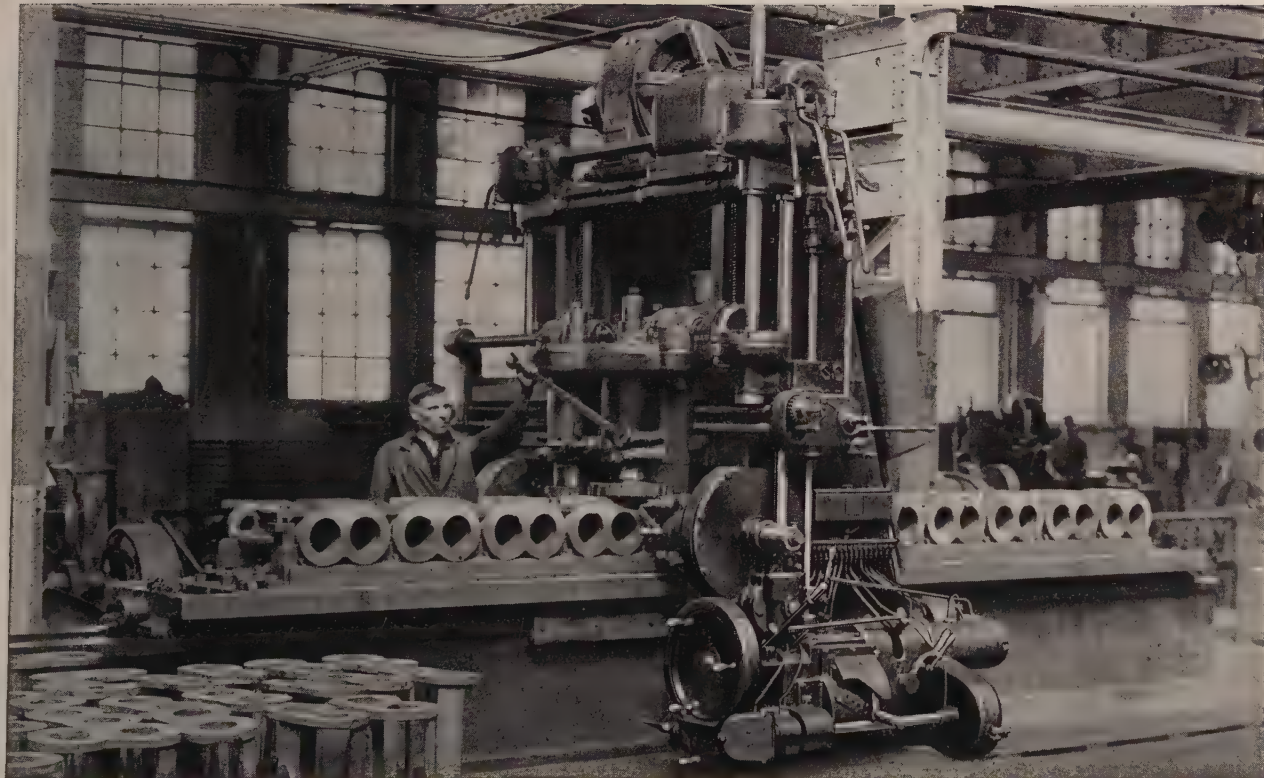


Fig. 39.

Accurate Machining to Insure Interchangeability.

is the largest ratio of cylinders, and the back pressure as 6 lb. per sq. in. Then

$$\text{m.e.p.} = 2 \times 120 - \frac{120}{4} - (6 \times 4)$$
$$= 186 \text{ lb. per sq. in.}$$

89. The area of the high-pressure cylinder would be

$$\frac{20{,}069}{186} = 107.9 \text{ sq. in.,}$$

and the low-pressure cylinder
$107.9 \times 4 = 431.6$ sq. in.
and the diameters $11\frac{3}{4}$ +, in. and $23\frac{3}{8}$ +, in. respectively.

The nearest standard would be 12-in. high-pressure and 22-in. low-pressure cylinder and the size of the condensing pump would be 12 and 22 by $10\frac{3}{4}$ by 18.

90. As **triple-expansion pumps** are nearly always operated condensing we will only consider the condensing pump with 26-in. vacuum and 5 lb. per sq. in. absolute back pressure.

$$R_1 = \sqrt[3]{\left(\frac{p}{p_b}\right)^2} = 8.32 = \sqrt[3]{\left(\frac{120}{5}\right)^2}$$

= Ratio low-pressure to high-pressure cylinder.

The practical limit for the ratio of areas is 8, hence

$$R_1 = 8 \text{ and } R_2 = \sqrt{8} = 2.83$$

$$\text{m.e.p.} = 3 \times 120 - \frac{2 \times 120}{2.83} - (5 \times 8)$$
$$= 235.2 \text{ lb. per sq. in.}$$

Area of high-pressure cylinder

$$= \frac{20{,}069}{235.2} = 85.2 \text{ sq. in., or } 10\tfrac{3}{8} +, \text{ diameter in inches.}$$

Area of intermediate-pressure cylinder
$= 85.2 \times 2.83$
$= 241.1$ sq. in., or $17\frac{1}{2}$ +, diameter in inches.

Area of low-pressure cylinder
$= 85.2 \times 8$
$= 681.6$ sq. in., or $29\frac{1}{2}$ −, diameter in inches.

The nearest commercial sizes of cylinders would be used, making the pump 12 and 19 and 30 by $10\frac{3}{4}$ by 18.

Fig. 40-41

Testing Worthington Pumps Before Shipment.

91. In commercial practice it may be advisable in many cases to lengthen the stroke and increase the piston speed to reduce the plunger size and consequently the size of the steam end. In this way a more economical operating unit may be obtained but one slightly more expensive in first cost.

92. For example, the triple-expansion pump in the preceding example may be made 24-in. stroke and the piston speed increased, which would enable us to use a $10\frac{1}{4}$-in. diameter plunger and reduce the size of the engine to 10 and 16 and 25 by $10\frac{1}{4}$ by 24.

93. TABLE OF STEAM FORCES

WORTHINGTON COMPOUND NON-CONDENSING STEAM ENDS

Steam Piston Force = Area of high-pressure Steam Piston × M.E.P.

$$\text{M.E.P.} = 2p - \frac{p}{R} - p_b R. \quad p = \text{Gage Pressure} + 10 \text{ lb. per sq. in}$$

$$p_b = 16 \text{ lb. per sq. in.}$$

Diameter Steam Cylinders Inches		Stroke Inches	$R = \frac{\text{Area L.P.}}{\text{Area H.P.}}$	Steam Pressure (lb. per sq. in. Gage)					
H.P.	L.P.			100	110	120	130	140	150
5¼	7½	10	2.04	2882	3208	3534	3861	4187	4513
6	9	10	2.25	3810	4249	4688	5126	5565	6004
8	12	12	2.25	6797	7579	8361	9144	9926	10708
9	14	12	2.42	8639	9648	10657	11667	12676	13685
10	16	12	2.56	10682	11945	13208	14472	15735	16998
12	18	12	2.25	15269	17027	18784	20542	22299	24057
14	20	12	2.04	20550	22875	25200	27525	29850	32175
9	14	15	2.42	8639	9648	10657	11667	12676	13685
12	17	15	2.01	15043	16741	18438	20136	21833	23531
14	20	15	2.04	20550	22875	25200	27525	29850	32175
10	16	18	2.56	10682	11945	13208	14472	15735	16998
12	18	18	2.25	15269	17027	18784	20542	22299	24057
14	20	18	2.04	20550	22875	25200	27525	29850	32175
14	22	18	2.47	20936	23393	25849	28306	30762	33219
16	25	18	2.44	27312	30508	33704	36901	40097	43293
18	29	18	2.59	34635	38748	42860	46973	51806	55199
10	16	24	2.56	10682	11945	13208	14472	15735	16998
12	18	24	2.25	15269	17027	18784	20542	22299	24057
14	20	24	2.04	20550	22875	25200	27525	29850	32175
14	22	24	2.47	20936	23393	25849	28306	30762	33219
16	25	24	2.44	27312	30508	33704	36901	40097	43293
19	30	24	2.49	38553	43085	47616	52148	56679	61211
21	34	24	2.62	47145	52751	58357	63964	69570	75176
23	38	24	2.73	56521	63309	70096	76884	83671	90459
25	42	24	2.83	66698	74785	82871	90958	99044	107131

94. TABLE OF STEAM FORCES

WORTHINGTON COMPOUND CONDENSING STEAM ENDS

Steam Force = Area of High Pressure Steam Piston × M.E.P.

$$\text{M.E.P.} = 2p - \frac{p}{R} - p_b R. \quad p = \text{Gage Pressure} + 10 \text{ lb. per sq. in.}$$

$$p_b = 6 \text{ lb. per sq. in.}$$

Diameter Steam Cylinders Inches		Stroke Inches	$R = \frac{\text{Area L.P.}}{\text{Area H.P.}}$	Steam Pressure (lb. per sq. in. Gage)					
H.P.	L.P.			100	110	120	130	140	150
9	16	15	3.16	10576	11647	12718	13789	14860	15931
12	20	15	2.78	18520	20375	22230	24086	25941	27796
15	25	15	2.78	28936	31834	34733	37631	40530	43428
10	18	18	3.24	13086	14413	15741	17068	18396	19723
12	22	18	3.36	18900	20825	22750	24676	26601	28526
14	25	18	3.19	25613	28213	30813	33414	36014	38614
16	29	18	3.29	33543	36953	40363	43773	47183	50593
10	18	24	3.24	13086	14413	15741	17068	18396	19723
15	25	24	2.78	28936	31834	34733	37631	40530	43428
16	30	24	3.52	33704	37154	40604	44055	47505	50955
21	36	24	2.94	57132	62881	68630	74379	80128	85877
21	38	24	3.27	57756	63623	69490	75358	81225	87092
23	42	24	3.33	69281	76363	83446	90528	97611	104693

95. TABLE OF STEAM FORCES

WORTHINGTON TRIPLE-EXPANSION STEAM ENDS

Steam Force = Area of high-pressure steam piston × M.E.P.

$$\text{M.E.P.} = 3\,p - 2\,\frac{p}{R_2} - p_b R_1 \quad p = \text{Gage Pressure} + 10 \text{ lb. per sq. in.}$$

p_b = 5 lb. per sq. in.

Diameter Steam Cylinders Inches			Stroke	$R_1 = \frac{\text{Area L.P.}}{\text{Area H.P.}}$	$R_2 = \frac{\text{Area I.P.}}{\text{Area H.P.}}$	Steam Pressure (lb. per sq. in. Gage)					
H.P.	I.P.	L.P.				100	110	120	130	140	150
6	9	16	15	7.12	2.25	5565	6160	6760	7355	7955	8550
8	12	20	15	6.25	2.25	10110	11170	12235	13295	14360	15420
9	14	22	18	5.98	2.42	13305	14685	16070	17450	18835	20215
10	16	25	18	6.25	2.52	16600	18330	20065	21795	23530	25260
12	18	29	18	5.84	2.25	22965	25350	27740	30125	32515	34900
10	16	25	24	6.25	2.52	16600	18330	20065	21795	23530	25260
12	19	30	24	6.25	2.50	23835	26325	28810	31300	33785	36275
13	21	34	24	6.85	2.62	28100	31040	33985	36925	39870	42810
15	23	38	24	6.43	2.35	36090	39885	43685	47480	51280	55075
16	25	42	24	6.89	2.44	41285	45665	50050	54430	58815	63195
18	29	46	24	6.53	2.60	54140	59820	65495	71175	76850	82530
19	30	50	24	6.93	2.49	58685	64890	71095	77300	83505	89710

100. Water Works Engines.—In water works practice the term "pumping engine" is applied to the large and highly efficient machines used for handling the large quantities of water required for this service.

101. Water-works pumps may be of the positive-displacement type or they may be of the centrifugal type, depending upon the conditions of service. The most used type is the duplex direct-acting pump; compound condensing for small units and triple-expansion condensing for large units. The crank-and-flywheel type with either compound or triple-expansion steam ends is used extensively for water-works service. These are highly efficient types of pumping engines and are very economical in operation. Recent developments in centrifugal-pump design have made this type of pump available for water-works service and many installations of both steam-turbine and motor-driven centrifugal pumps have been made by Worthington. Power-driven displacement pumps are used to some extent and for small installations show very good economy.

102. In large industrial plants, oil refineries, etc., the steam consumption of the pumping machinery is not of much importance, especially when the exhaust steam can be utilized for feed-water heating, processing, etc. But where the entire plant is a pumping operation only, like a water-works plant, **steam economy** is of great importance.

103. In speaking of the economy of a pumping engine, the term **"duty"** is always used, and may be defined as follows: "Duty" is the number of foot-pounds of useful work done by one thousand pounds of dry steam, or the number of foot-pounds of useful work done by one million B.t.u.

104. The term "duty" originally referred to the amount of work done by 100 lb. of coal. This method introduced so many variables, such as efficiency of boiler, quality of the coal and skill in firing, that its use was abandoned. "Duty" was next expressed as the foot-pounds of work done per 1000 lb. of dry steam. This method is closely related to the 100 lb. of coal basis of computation, as a high-grade boiler will evaporate 10 lb. of water per pound of coal and 100 lb. of coal will produce 1000 lb. of dry steam.

105. In 1891, a committee of the **A.S.M.E.** reported on a **Standard** method of conducting duty trials. This committee also recommended that a new unit be used—viz. foot-pounds of useful work done for one million B.t.u. furnished by the boiler instead of foot-pounds of useful work done based on 1000 lb. of dry steam.

106. A comparison of the two units based on steam at 150 lb. per sq. in. gage pressure (165 lb. per sq. in. absolute) shows:

Total heat above zero	=	1227 B.t.u.
Feedwater temperature, say 120°	=	120 "
Difference	=	1107 "

107. Therefore, one million B.t.u. will evaporate $\frac{1,000,000}{1107}$ or 905 lb. of water as against the old unit of 1000 lb. of water evaporated per 100 lb. of coal. The duty based on the new unit will be about 10 per cent less than the duty based on the old unit. A manufacturer who bases his guarantee on the new unit would apparently be bidding on an engine of lower economy than a manufacturer who bases his duty on the old unit, and a true comparison could only be made by recalculating the duties.

108. Two **factors are required** to be known for **calculating the duty** of a pumping engine. These are

1. The work done $= W$ foot-pounds
2. The steam consumed $= S$ pounds

The duty is then $\frac{1000\ W}{S}$ ft-lb. per 1000 lb. steam. This is the old method. The duty can then be expressed in the new unit of foot-pounds of work done per million B.t.u. by finding the B.t.u. in the 1000 lb. of steam consumed.

109. The quantity W is found by **measuring** the amount of **water** pumped in pounds and the height to which it has been pumped, (total dynamic head). The quantity of water may be measured by a meter or by a weir. If the engine is of the positive-displacement type and in good condition, the quantity pumped can be determined by calculating the displacement of the plungers and deducting for slip and rods. This will not introduce any more serious error than the error that may be expected in any other method of measuring.

110. The total dynamic head is measured as described in Sec. I, par. 43 to par. 46. The temperature of the water is also noted and used in calculating the capacity in pounds and the head in feet.

111. The **steam consumption** S can be **measured** in two ways that are accurate and at least two that are approximate. The two accurate ways are:

1—Condense and weigh all the steam that passes through the steam end of the pump.

2—Run the pump from one boiler and measure all the feed-water taken by this boiler during the time of the test. It is essential that the water level in the boiler be the same at the end of the test as at the start.

112. Of the approximate methods the first is to measure the steam by a "steam meter" in the line running to the pump; the second is to measure it by the indicator diagrams.

113. Water-Works Surface Steam Condensers.—As the subject of condensers is closely related to water-works pumping engines, a few words on this important auxiliary may be permissible.

114. The surface condenser is used almost exclusively in water-works practice. The **water-works type differs** from the regular type in that the cooling water flows around the outside of the tubes, and the steam passes through the tubes. These condensers are nearly always built with one pass for both the steam and the water.

115. In water-works practice it is customary to **locate the surface condenser** in either the suction or discharge line of the pumping engine in order to utilize the water pumped by the main engine for circulating water in the condenser. In the majority of cases, the condenser is placed in the suction. When the suction lift is excessive or when space conditions prohibit, the condenser is placed in the discharge line.

116. In **designing water-works condensers,** allowance must be made for ample water ways around the tubes, and ample steam flow area within the tubes. These points are ofttimes overlooked by inexperienced designers. The excessive friction resulting from this oversight will reduce the capacity of the main pumping

unit if the condenser is located in the suction line and will add to the discharge head if the condenser is located in the discharge line.

117. The **condenser shells** are usually of cylindrical form, and fitted with brass tubes passing through stuffing boxes, located in cast iron or **Muntz** metal tube heads as shown by Fig. 42.

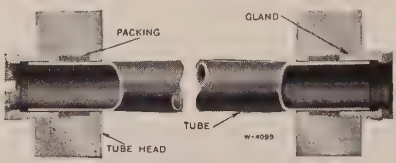


Fig. 42. Condenser tube, and tube heads with stuffing boxes and glands.

118. The **tubes heads** are carefully drilled and tapped to receive the threaded **b.onze glands** which are provided with a lip on the inside to prevent the tube from creeping. This design allows for freedom of expansion and contraction, and for the quick removal or repacking of any tube in case of leakage. The stuffing boxes are packed with lacing, fibre ferrules or wicking, as may be preferred.

119. The main exhaust pipe is connected to the **exhaust head** at one end of the condenser and the **air-pump** suction to the **crown head** at the other end.

120. Fig. 43 and Fig. 44 show respectively surface condensers in the suction and **discharge lines** of water-works engines.

121. The rate of **transfer of heat** through the condensing surface varies with conditions. In some localities lime is deposited on the tubes, reducing the heat transfer. In others mud and other foreign materials are encountered, with the same results. In a water-works condenser much of the water flows by the tubes with very little, if any, cooling effect. We, therefore, base our calculations for water-works condensers on a heat transfer of 250 B.t.u. per sq. ft. per hour per degree of mean temperature difference.

122. The **amount of cooling surface** in square feet **necessary** to condense a given amount of steam can be calculated from the formula

$$F = \frac{W H}{T U}$$

where F is the square feet cooling surface; W the weight of exhaust steam per hour; H the B.t.u. per pound of exhaust steam; T the mean temperature difference between the circulating water

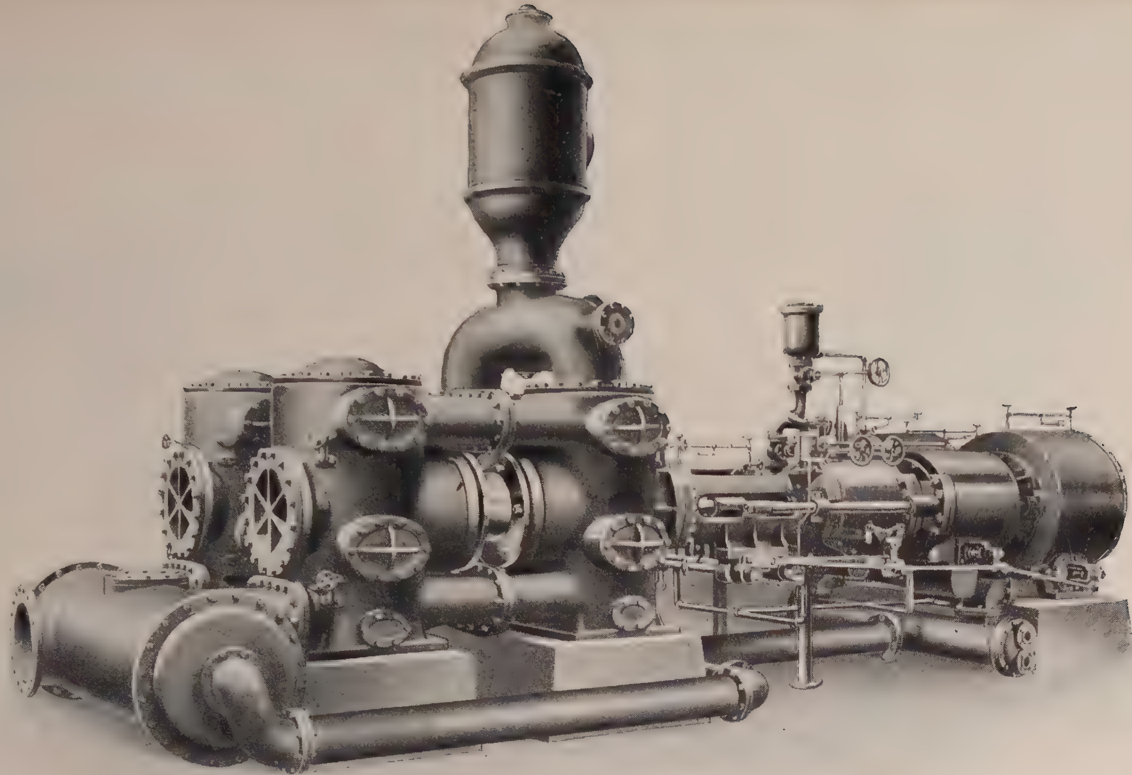


Fig. 43.
Worthington Horizontal Triple-expansion Water-works Pumping Engine with Surface Condenser in the Suction.

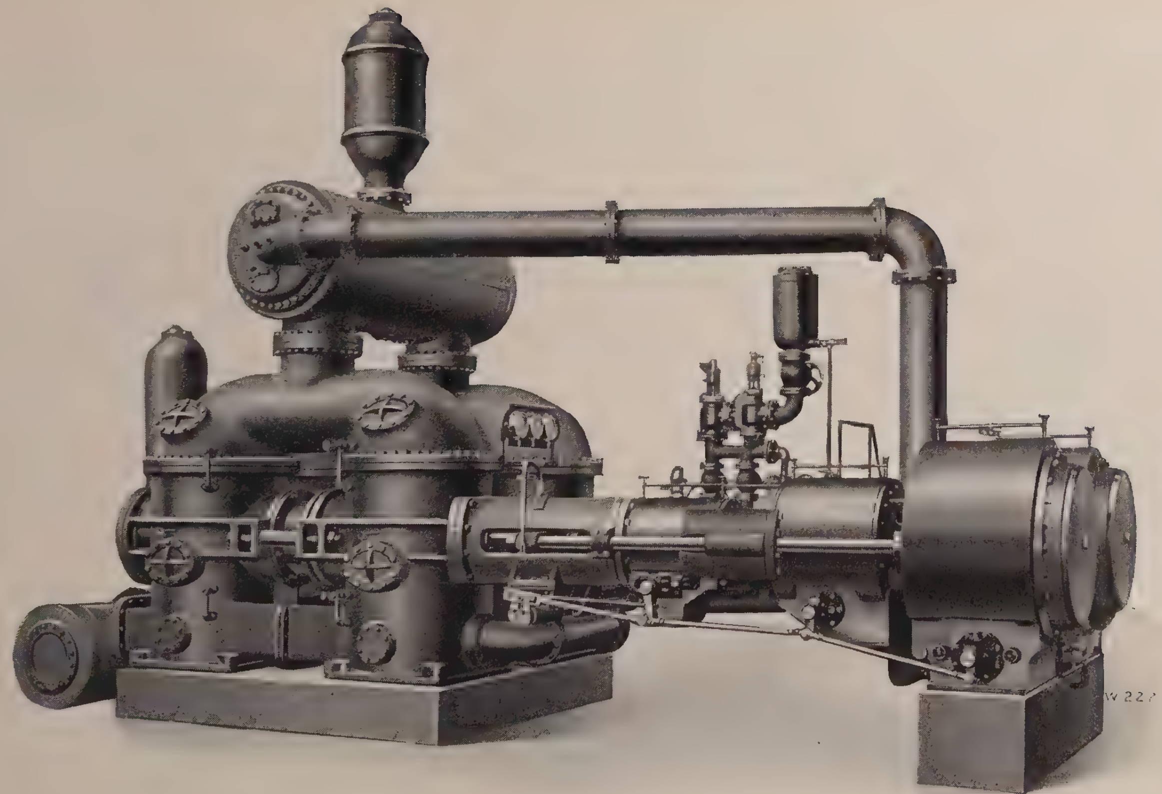


FIG. 44.
WORTHINGTON HORIZONTAL TRIPLE-EXPANSION WATER-WORKS PUMPING ENGINE WITH SURFACE CONDENSER IN THE DISCHARGE

and the exhaust steam, and U the rate of heat transfer in B.t.u. per 1 deg. mean difference per sq. ft. of condensing surface per hour. The B.t.u. per lb. of exhaust steam per hour (H) is frequently assumed to be:

930 at 4 in. absolute; 935 at 3 in. absolute; 940 at 2 in. absolute; 945 at 1 in. absolute.

123. In commercial calculations, a value of 950 B.t.u. is taken as approximately correct for H. The value of T is found from the formula

$$T = T_s - \frac{T_o + T_1}{2}$$

where T_s is the temperature in steam space (assumed to correspond to absolute pressure (Sec. I, Table 87); T_1 the temperature of circulating water at condenser inlet in deg. F., and T_o the temperature of circulating water at condenser outlet in degrees F.

124. For vacuum of 26 in. and under, the **wet vacuum pump** is generally used. This pump will remove both the condensate, and the non-condensable vapors. The size of the wet vacuum pump is not so easily calculated, as the displacement of this pump depends entirely upon the amount of air to be removed, and not upon the amount of steam. A direct-acting pump is generally used for this service, but on account of its large cylinder clearances it is considered good practice to make the capacity of the wet vacuum pump for a surface condenser equal to 20 to 30 times the pounds of steam condensed per unit of time. The following formula may serve as a guide to the multiple to use:

$$DV = S\,1.5\left(\frac{54}{P_a} + 1\right)$$

where DV is the displacement of vacuum pump in pounds per hour; S the pounds of steam per hour; P_a the absolute pressure in inches of mercury, and $\frac{54}{P_a}$ the amount of vapor arising from the steam.

125. For high vacuum condensers **dry vacuum pumps** are necessary for removing the air and non-condensable vapors. The Worthington crank-and-flywheel type of vacuum pump, with close clearances in the vacuum cylinders and high volumetric efficiency, is the

type best adapted for this purpose. The displacement of the dry vacuum pump

$$= \frac{54}{P_a} S, \text{ approximately.}$$

126. When a dry vacuum pump is used, the condensate is removed from the condenser by a separate pump called the **hotwell pump,** usually a two-stage centrifugal. Direct-acting pumps may also be used.

127. The **displacement** of the hotwell pump is equal to the pounds of condensate, thus

$$\frac{\text{pounds steam per hour}}{8.33 \times 60} = \text{gal. per min.}$$

Example; 5000 lb. steam per hour; 26 in. vacuum; 70 deg. circulating water inlet

$$T = 125.48 - \frac{110.48 + 70}{2} = 35.24$$

$$F = \frac{5000 \times 950}{35.24 \times 250} = 538 \text{ sq. ft.}$$

Displacement of wet vacuum pump

$$DV = 5000 \times 1.5 \left(\frac{54}{4} + 1\right) = 5000 \times 21.75 = 108{,}750 \text{ lb.}$$

Therefore

$$\frac{108{,}750}{8.33 \times 60} = 218 \text{ gal. per min.}$$

128. We would select for this service an 8 by 10 by 12 horizontal simplex wet-vacuum pump operating at a piston speed of 54 ft. per min.

129. For high-vacuum work the selection of the proper size condenser, dry-vacuum pump and hotwell pump should be left to the manufacturer.

130. As water-works pumping engines are always special and built to suit one given set of conditions, no attempt can be made to list sizes and capacities.

131. Typical water-works installations by Worthington are shown by Fig. 43 and Fig. 44.

WORTHINGTON GENERAL SERVICE PUMPS
Piston Pattern

140. General service pumps are suitable for a wide range of service. They are used for pumping water (hot or cold), acids, alkalines, fuel oil, gasoline, kerosene and other light distillates, for boiler feeding, etc. where medium discharge pressures are encountered. Worthington Duplex General Service Pumps, piston-pattern, are available for the capacities and pressures listed below. A complete line of each type will be found in the following tables.

Gal. per Min. Water and Similar Liquids	b. hp. Boiler Feeding	Max. Disch. Press.	Type of Liquid End	Table No.
9–165	80–1515	200	Valve Plate	141
20–165	180–1515	150	Removable Liner	142
225–500	2050–4700	150	Turret	143–144
265–475	2500–4450	150	Oblong Force Chamber	145
170–685	1575–6450	250	Separate Cylinder	146–147
170–930	....	150	Straightway	148–149

NOTE—Worthington carries a complete line of patterns for pumps of greater capacities than found in the above list. For pumps of greater capacities and for pumps to meet special conditions of service consult the nearest Worthington branch office.

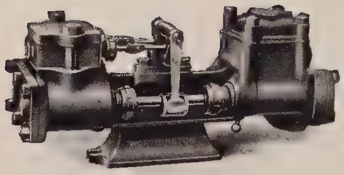


Fig. 45.

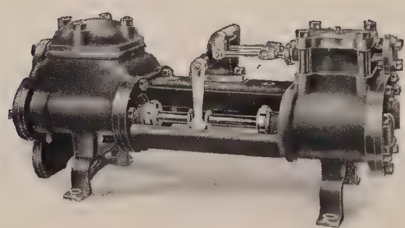


Fig. 46

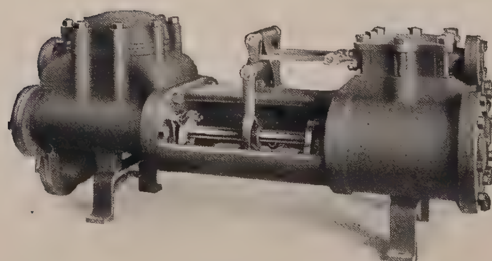


Fig. 47.

Worthington Duplex Packed-Piston Pumps

Valve-Plate Type

WORTHINGTON DUPLEX PACKED-PISTON PUMPS

Valve-Plate Type

Maximum working pressure: Steam end—150 lb.
Liquid end—200 lb.

141. TABLE OF SIZES AND DATA

Fig. No.	Pipe Sizes, Inches: Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. of Boiler Pump Will Feed	Size of Pump in Inches: Steam	Exhaust	Suction	Discharge	Floor Space Inches
45	3	2	3	200	9	28	55	80	⅜	½	1¼	1	24 x 9
45	3½	2¼	4	200	13	33	50	120	⅜	¾	1½	1¼	28 x 9
47	4½	2¾	4	200	20	33	50	180	½	¾	2	1½	33 x 13
46	5¼	3½	5	200	36	38	45	325	¾	1¼	2½	1½	38 x 16
46	6	4	6	200	51	40	40	460	1	1¼	3	2	44 x 17
46	7½	5	6	200	80	40	40	725	1½	2	4	3	45 x 22
46	7½	4½	10	200	94	58	35	845	1½	2	4	3	59 x 22
46	9	5¼	10	200	126	58	35	1170	2	2½	4	3	61 x 22
47	10	6	10	200	165	58	35	1515	2	2½	5	4	62 x 36

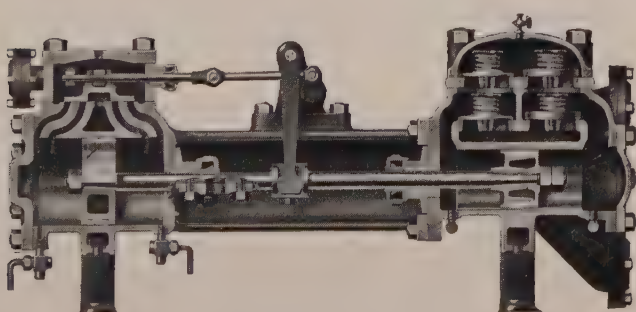


Fig. 48. Sectional view of Standard Duplex Piston Pump, Vale-plate Type.

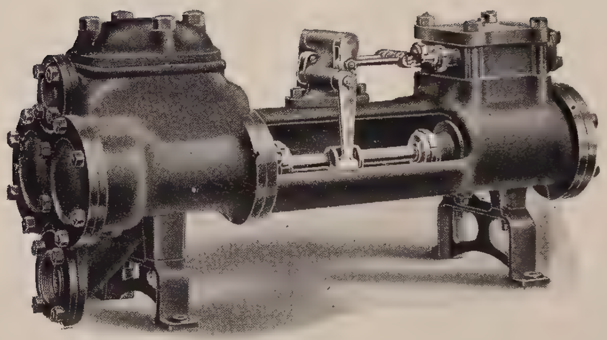


Fig. 49

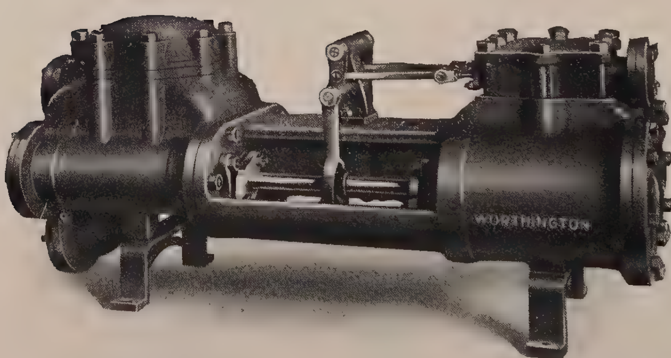


Fig. 50.

Worthington Duplex Packed-Piston Pumps

Valve-Plate Type with Removable Liners

WORTHINGTON DUPLEX PACKED-PISTON PUMPS

Valve-Plate Type with Removable Liners

Maximum working pressure: Steam end—150 lb.
Liquid end—150 lb.

142. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. of Boiler Pump Will Feed	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space Inches
49	4½	2¾	4	150	20	33	50	180	½	¾	2	1½	33 x 13
49	5¼	3½	5	150	36	38	45	325	¾	1¼	2½	1½	38 x 16
49	6	4	6	150	51	40	40	460	1	1¼	3	2	45 x 17
49	7½	5	6	150	80	40	40	725	1½	2	4	3	45 x 22
49	7½	4½	10	150	94	58	35	845	1½	2	4	3	59 x 22
50	9	5¼	10	150	126	58	35	1170	2	2½	4	3	61 x 23
50	10	6	10	150	165	58	35	1515	2	2½	5	4	62 x 26

WORTHINGTON DUPLEX PACKED-PISTON PUMPS

Turret Pattern

Maximum working pressure:
Steam end—150 lb.
Liquid end—See table.

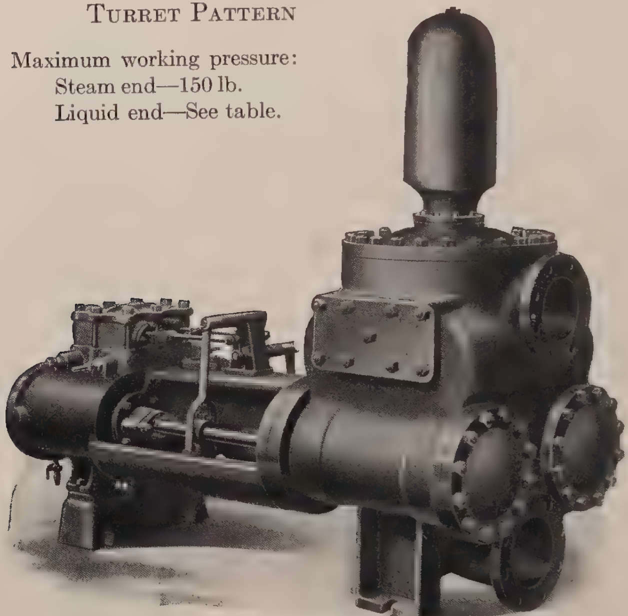


Fig. 51

143. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boiler Pump Will Feed	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space Inches
51	10	7	10	200	225	58	35	2050	2	2½	6	5	60 x 26
51	12	7	12	175	233	60	30	2100	2	2½	6	5	76 x 31
51	14	7	12	175	233	60	30	2100	2½	3	6	5	78 x 33
51	12	8½	12	175	344	60	30	3200	2	2½	6	6	76 x 31
51	14	8½	12	175	344	60	30	3200	2½	3	6	6	78 x 33
51	16	8½	12	175	344	60	30	3200	2½	3	6	6	79 x 40
51	14	10¼	12	175	500	60	30	4700	2½	2½	8	7	78 x 33
51	16	10¼	12	175	500	60	30	4700	2½	3	8	7	79 x 40

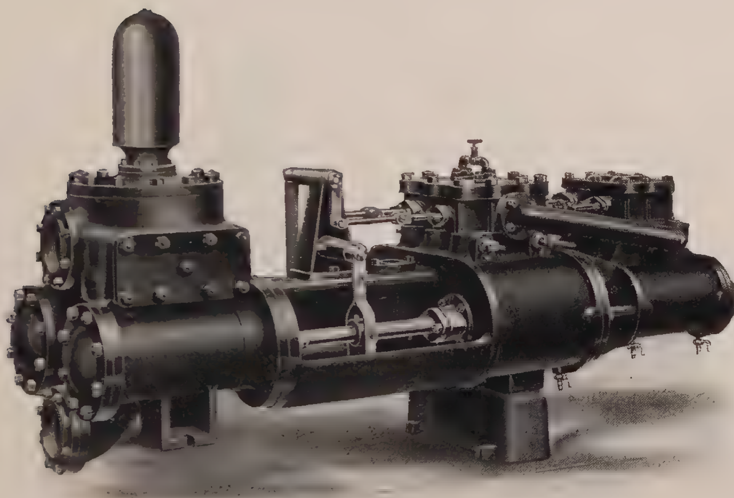


FIG. 52

WORTHINGTON COMPOUND DUPLEX PACKED-PISTON PUMPS

TURRET PATTERN

Maximum working pressure: Steam end—150 lb.
Liquid end—See table.

144. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boiler Pump Will Feed	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space Inches
52	8 & 12	7	12	175	233	60	30	2100	2	3	6	6	87 x 36
52	9 & 14	7	12	175	233	60	30	2100	2	3	6	6	89 x 37
52	8 & 12	8½	12	175	344	60	30	3200	2	3	6	6	87 x 36
52	9 & 14	8½	12	175	344	60	30	3200	2	3	6	6	89 x 37
52	9 & 14	10¼	12	175	500	60	30	4700	2	3	8	7	92 x 40
52	10 & 16	10¼	12	175	500	60	30	4700	2	3	8	7	92 x 40

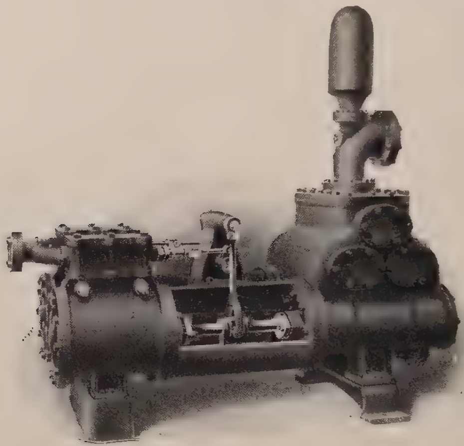


FIG. 53.

WORTHINGTON DUPLEX PACKED-PISTON PUMPS

OBLONG-FORCE-CHAMBER TYPE

Maximum working pressure: Steam end—150 lb.
Liquid end—150 lb.

145. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke	Max. Discharge Pressure Good For	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boiler Pump Will Feed	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space, Inches
53	12	7½	12	150	267	60	30	2500	2½	3	6	6	80 x 36
53	14	7½	12	150	267	60	30	2500	2½	3	6	6	80 x 37
53	16	7½	12	150	267	60	30	2500	2½	3	6	6	82 x 46
53	12	8½	12	150	344	60	30	3200	2½	3	6	6	83 x 36
53	14	8½	12	150	344	60	30	3200	2½	3	6	6	83 x 37
53	16	8½	12	150	344	60	30	3200	2½	3	6	6	86 x 46
53	14	10	12	150	475	60	30	4450	2½	3	8	7	83 x 37
53	16	10	12	150	475	60	30	4450	2½	3	8	7	86 x 46
53	18	10	12	150	475	60	30	4450	3	3½	8	7	86 x 47
53	20	10	12	150	475	60	30	4450	4	5	8	7	86 x 50

WORTHINGTON HEAVY PATTERN DUPLEX PACKED-PISTON PUMPS

Maximum working pressure:
Steam end—150 lb.
Liquid end—250 lb.

FIG. 54.

146. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke	Max. Discharge Pressure Good For	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boiler Pump Will Feed	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space, Inches
54	8	6	12	250	171	60	30	1575	1½	2	5	4	75 x 30
54	10	6	12	250	171	60	30	1575	2	2½	5	4	75 x 30
54	10	7	12	250	233	60	30	2100	2	2½	6	5	77 x 39
54	12	7	12	250	233	60	30	2100	2½	3	6	5	83 x 39
54	14	7	12	250	233	60	30	2100	2½	3	6	5	83 x 39
54	12	8½	12	250	344	60	30	3200	2½	3	8	6	86 x 40
54	14	8½	12	250	344	60	30	3200	2½	3	8	6	87 x 40
54	16	8½	12	250	344	60	30	3200	3	4	8	6	87 x 46
54	18	8½	12	250	344	60	30	3200	3	4	8	6	88 x 48
54	20	8½	12	250	344	60	30	3200	4	5	8	6	88 x 50
54	16	10	12	250	475	60	30	4450	3	4	8	7	90 x 48
54	18	10	12	250	475	60	30	4450	3	4	8	7	91 x 48
54	20	10	12	250	475	60	30	4450	4	5	8	7	91 x 50
54	16	12	12	250	685	60	30	6450	3	4	10	8	92 x 49
54	18	12	12	250	685	60	30	6450	3	4	10	8	92 x 49
54	20	12	12	250	685	60	30	6450	4	5	10	8	92 x 50

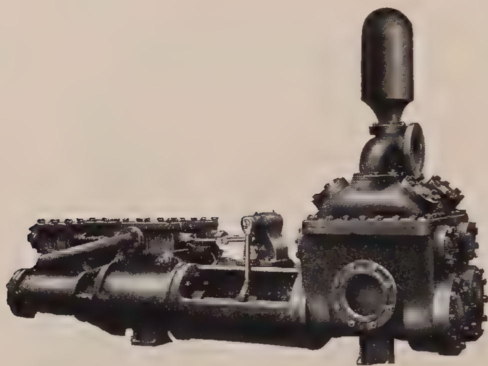


Fig. 55

WORTHINGTON COMPOUND DUPLEX PACKED-PISTON PUMPS

Heavy Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end—250 lb.

147. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation				Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boiler Pump Will Feed	Steam	Exhaust	Suction	Discharge	
55	8 & 12	7	12	250	233	60	30	2100	2	3	6	5	107 x 39
55	9 & 14	7	12	250	233	60	30	2100	2	3	6	5	108 x 39
55	8 & 12	8½	12	250	344	60	30	3200	2	3	8	6	110 x 46
55	9 & 14	8½	12	250	344	60	30	3200	2	3	8	6	111 x 46
55	10 & 16	8½	12	250	344	60	30	3200	2	3	8	6	112 x 46
55	12 & 18	8½	12	250	344	60	30	3200	2½	3½	8	6	113 x 48
55	14 & 20	8½	12	250	344	60	30	3200	2½	5	8	6	114 x 50
55	10 & 16	10	12	250	475	60	30	4450	2	3	8	7	115 x 48
55	12 & 18	10	12	250	475	60	30	4450	2½	3½	8	7	116 x 48
55	14 & 20	10	12	250	475	60	30	4450	2½	5	8	7	117 x 50
55	10 & 16	12	12	250	685	60	30	6450	2	3	10	8	117 x 49
55	12 & 18	12	12	250	685	60	30	6450	2½	3½	10	8	118 x 49
55	14 & 20	12	12	250	685	60	30	6450	2½	5	10	8	119 x 50

FIG. 56.

(For table of sizes see following page)

WORTHINGTON DUPLEX PACKED-PISTON PUMP

Straightway Pattern

WORTHINGTON DUPLEX PACKED-PISTON PUMPS

Straightway Pattern

Fig. 56. See Page 55.

Maximum working pressure: Steam end—150 lb.
Liquid end—150 lb.

148. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke	Max. Discharge Pressure Good For	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space Inches
56	10	6	12	150	171	60	30	2	2½	5	4	77 x 30
56	8	7	12	150	233	60	30	1½	2	6	5	76 x 34
56	10	7	12	150	233	60	30	2	2½	6	5	77 x 34
56	12	7	12	150	233	60	30	2½	3	6	5	83 x 36
56	14	7	12	150	233	60	30	2½	3	6	5	83 x 37
56	10	8½	12	150	344	60	30	2	2½	6	5	78 x 43
56	12	8½	12	150	344	60	30	2½	3	6	5	84 x 43
56	14	8½	12	150	344	60	30	2½	3	6	5	84 x 43
56	16	8½	12	150	344	60	30	2½	3	6	5	84 x 49
56	12	10	12	150	475	60	30	2½	3	8	7	88 x 46
56	14	10	12	150	475	60	30	2½	3	8	7	89 x 46
56	16	10	12	150	475	60	30	2½	3	8	7	91 x 52
56	18	10	12	150	475	60	30	3	3½	8	7	91 x 52
56	20	10	12	150	475	60	30	4	5	8	7	91 x 52
56	14	12	12	150	685	60	30	2½	3	10	8	90 x 51
56	16	12	12	150	685	60	30	2½	3	10	8	89 x 50
56	18	12	12	150	685	60	30	3	3½	10	8	91 x 49
56	20	12	12	150	685	60	30	4	5	10	8	91 x 50
56	16	14	12	150	930	60	30	2½	3	12	10	94 x 53
56	18	14	12	150	930	60	30	3	3½	12	10	94 x 53
56	20	14	12	150	930	60	30	4	5	12	10	94 x 53
56	14	10	15	150	594	75	30	2½	3	8	7	99 x 46
56	17	10	15	150	594	75	30	2½	3½	8	7	99 x 46
56	20	10	15	150	594	75	30	4	5	8	7	99 x 46
56	17	11	15	150	718	75	30	2½	3½	10	8	99 x 52
56	20	11	15	150	718	75	30	4	4	10	8	99 x 55
56	20	12	15	150	855	75	30	5	5	12	10	104 x 55

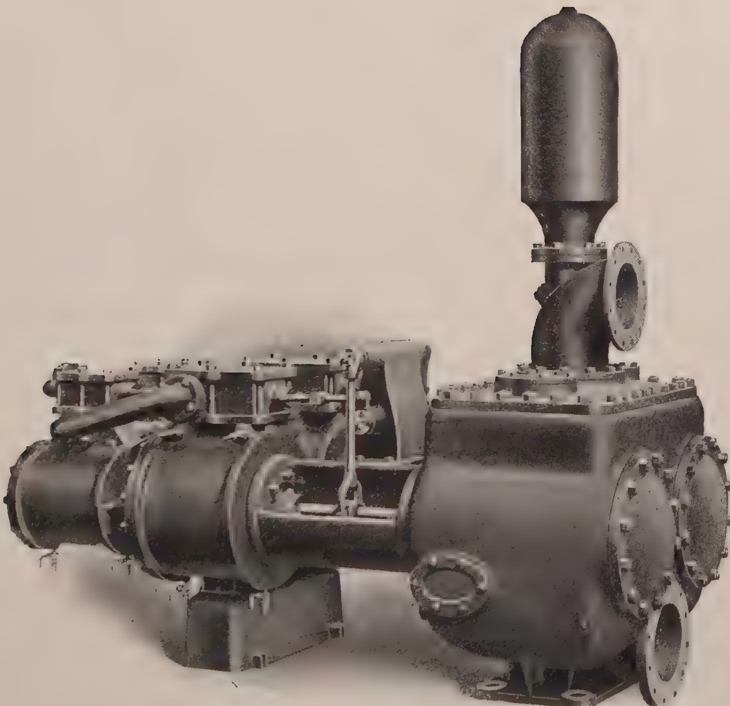


Fig. 57.

(For table of sizes see following page)

Worthington Compound Duplex Packed-Piston Pump

Straightway Pattern

WORTHINGTON COMPOUND DUPLEX PACKED-PISTON PUMPS

Straightway Pattern

Fig 57. See Page 57.

Maximum working pressure: Steam end—150 lb.
Liquid end—150 lb.

149. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke	Max. Discharge Pressure Good for	Capacity For Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space Inches
57	8 & 12	7	12	150	233	60	30	2	3	6	5	107 x 36
57	9 & 14	7	12	150	233	60	30	2	3	6	5	108 x 39
57	10 & 16	7	12	150	233	60	30	2	3	6	5	110 x 46
57	12 & 18	7	12	150	233	60	30	2½	3½	6	5	111 x 48
57	9 & 14	8½	12	150	344	60	30	2	3	6	5	109 x 43
57	10 & 16	8½	12	150	344	60	30	2	3	6	5	110 x 46
57	12 & 18	8½	12	150	344	60	30	2½	3½	6	5	111 x 48
57	14 & 20	8½	12	150	344	60	30	2½	5	6	5	112 x 50
57	10 & 16	10	12	150	475	60	30	2	3	8	7	115 x 46
57	12 & 18	10	12	150	475	60	30	2½	3½	8	7	117 x 48
57	14 & 20	10	12	150	475	60	30	2½	5	8	7	117 x 50
57	12 & 18	12	12	150	685	60	30	2½	3½	10	8	117 x 50
57	14 & 20	12	12	150	685	60	30	2½	5	10	8	117 x 50
57	14 & 20	14	12	150	930	60	30	2½	5	12	10	120 x 53
57	9 & 14	10	15	150	594	75	30	2	3	10	8	137 x 49
57	12 & 17	10	15	150	594	75	30	2½	3½	8	7	143 x 49
57	14 & 20	10	15	150	594	75	30	2½	5	8	7	143 x 49
57	9 & 14	11	15	150	718	75	30	2	3	10	8	137 x 49
57	12 & 17	11	15	150	713	75	30	2½	3½	10	8	143 x 49
57	14 & 20	11	15	150	718	75	30	2½	5	10	8	143 x 50
57	14 & 20	12	15	150	855	75	30	2½	5	12	10	143 x 52

WORTHINGTON LOW-SERVICE PUMPS
PISTON PATTERN

160. Low-service pumps are designed for discharge pressures up to 75 lb. per sq. in. The ratio of steam to liquid ends is such that the pumps may operate with the low steam pressures frequently encountered in small isolated pumping plants. Low-service pumps are used for tank supply, etc., where the discharge head does not exceed 175 feet (75 lb. per sq. in.). They are suitable for water, acids, alkalines, gasoline and light distillates. Worthington duplex low-service piston pattern pumps are available for the capacities and pressures listed below. A complete line of each type will be found in the tables indicated.

Gal. per min. Water and Similar Liquids	Max. Disch. Pressure Lb. per sq. in.	Type of Liquid End	Table No.
15– 149	75	Valve Plate	161
200– 840	75	Turret	162–163
300–1550	75	Straightway	164-165
950–2600	40–50	Separate Valve Chest	166–167

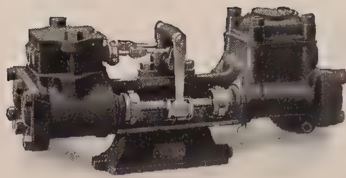


FIG. 58.

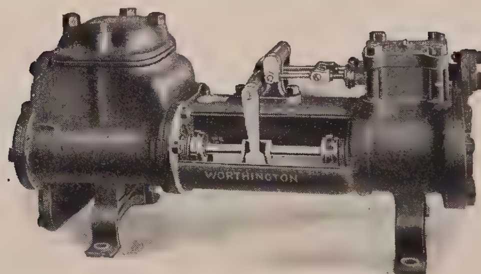


FIG. 59.

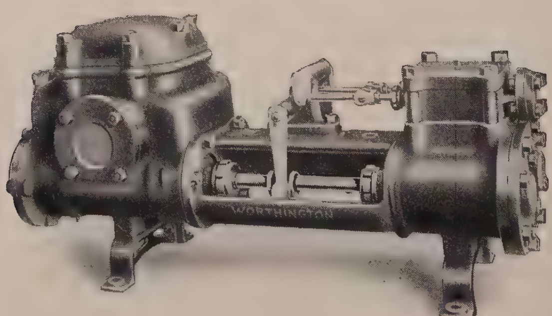


FIG. 60.

WORTHINGTON DUPLEX PISTON PUMPS

Valve-Plate Type

FOR LOW-PRESSURE SERVICE

WORTHINGTON DUPLEX PISTON PUMPS

Valve-Plate Type

Maximum working pressure: Steam end—150 lb.
Liquid end— 75 lb.

161. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation			Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
58	3	2¾	3	75	15	25	50	⅜	½	1¼	1	24 x 9
59	4½	3¾	4	75	33	30	45	½	¾	2½	1½	35 x 13
59	5¼	4¾	5	75	60	34	41	¾	1¼	3	2	39 x 16
60	6	5¾	6	75	94	36	36	1	1¼	4	3	47 x 17
60	7½	7½	6	75	160	36	36	1½	2	6	5	49 x 22
60	7½	8½	6	75	206	36	36	1½	2	6	5	49 x 22
59	7½	6	10	75	149	52	32	1½	2	5	4	57 x 30

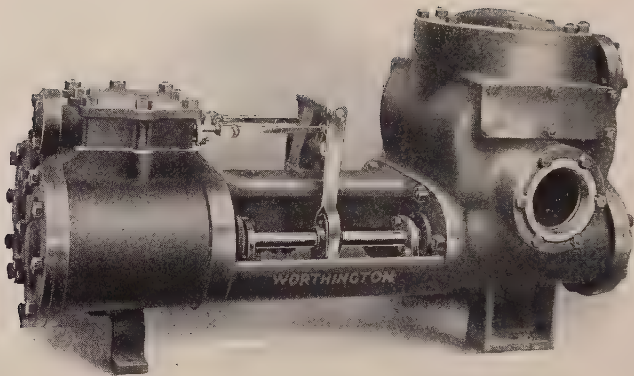


FIG. 61.

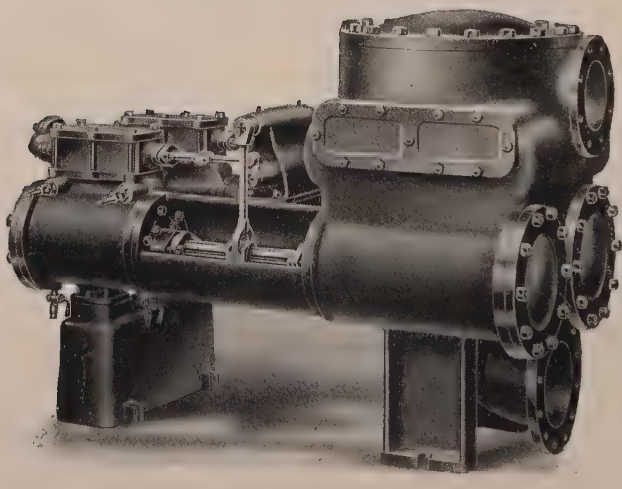


FIG. 62.

WORTHINGTON DUPLEX PISTON PUMPS

Turret Pattern

FOR LOW-PRESSURE SERVICE

WORTHINGTON DUPLEX PISTON PUMPS

Turret Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end— 75 lb.

162. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation			Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
61	7½	7	10	75	202	52	32	1½	2	6	5	58 x 24
61	7½	8½	10	75	296	52	32	1½	2	6	5	59 x 24
61	9	8½	10	75	296	52	32	2	2½	6	5	61 x 24
61	8	8½	12	75	308	54	27	1½	2	6	5	65 x 24
61	10	10¼	12	75	448	54	27	2	2½	8	7	72 x 36
62	10	12	12	75	615	54	27	2	2½	10	8	78 x 36
62	12	12	12	75	615	54	27	2½	3	10	8	84 x 36
	12	14	12	75	840	54	27	2½	3	12	10	88 x 42
	14	14	12	75	840	54	27	2½	3	12	10	88 x 42

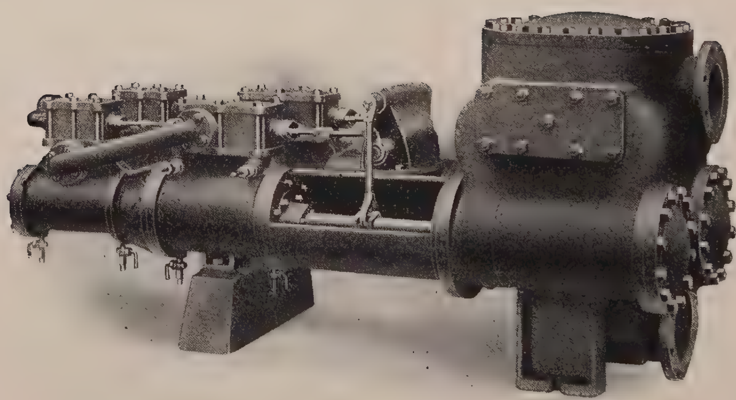


Fig. 63.

Worthington Compound Duplex Piston Pump

Turret Pattern

For Low-pressure Service

WORTHINGTON COMPOUND DUPLEX PISTON PUMPS

Turret Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end— 75 lb.

163. TABLES OF SIZES AND DATA—Continued

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation			Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
63	5¼ & 7½	7	10	75	202	52	32	1¼	2	6	5	77 x 22
63	6 & 9	7	10	75	202	52	32	1¼	2½	6	5	80 x 23
63	5¼ & 7½	8½	10	75	296	52	32	1¼	2	6	5	78 x 24
63	6 & 9	8½	10	75	296	52	32	1¼	2½	6	5	81 x 24
	9 & 14	10¼	12	75	448	54	27	2	3	8	7	107 x 39
	8 & 12	12	12	75	615	54	27	2	3	10	8	108 x 36
	9 & 14	12	12	75	615	54	27	2	3	10	8	109 x 39
	8 & 12	14	12	75	840	54	27	2	3	12	10	112 x 42
	9 & 14	14	12	75	840	54	27	2	3	12	10	113 x 42

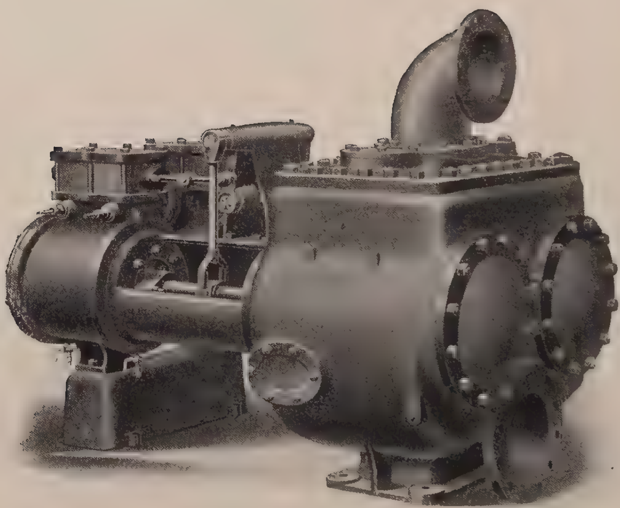


Fig. 64.

Worthington Duplex Packed-Piston Pump

Straightaway Pattern

For Low-pressure Service

WORTHINGTON DUPLEX PACKED-PISTON PUMPS

Straightway Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end— 75 lb.

164. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation			Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
64	8	8½	12	75	308	54	27	1½	2	6	5	77 x 43
64	8	10	12	75	427	54	27	1½	2	8	7	82 x 46
64	10	12	12	75	615	54	27	2	2½	10	8	83 x 49
64	12	12	12	75	615	54	27	2½	3	10	8	89 x 49
64	10	14	12	75	840	54	27	2	2½	12	10	86 x 53
64	12	14	12	75	840	54	27	2½	3	12	10	92 x 53
64	14	14	12	75	840	54	27	2½	3	12	10	92 x 53
64	12	16	12	75	1090	54	27	2½	3	14	12	102 x 55
64	14	16	12	75	1090	54	27	2½	3	14	12	102 x 55
64	16	16	12	75	1090	54	27	2½	3	14	12	104 x 55
64	12	15	15	75	1210	68	27	2½	3	14	12	107 x 55
64	14	15	15	75	1210	68	27	2½	3	14	12	112 x 55
64	17	15	15	75	1210	68	27	2½	3½	14	12	112 x 55
64	17	17	15	75	1550	68	27	2½	3½	14	12	112 x 55
64	20	17	15	75	1550	68	27	4	5	14	12	112 x 55

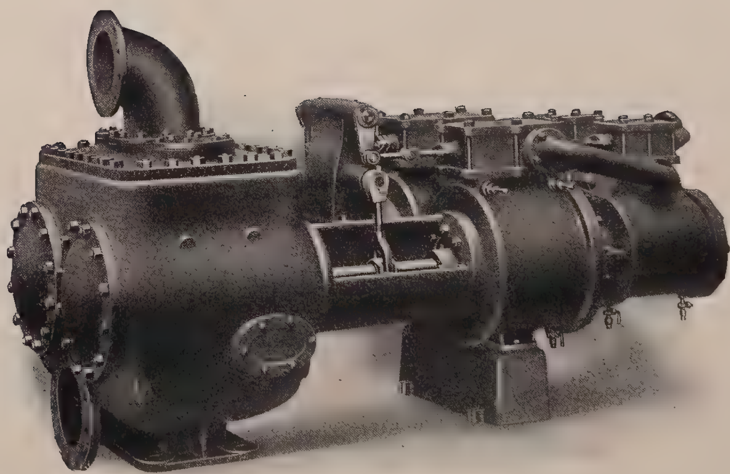


Fig. 65.

Worthington Compound Duplex Packed-Piston Pump

Straightway Pattern

For Low-pressure Service

WORTHINGTON COMPOUND DUPLEX PACKED-PISTON PUMPS

Straightway Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end— 75 lb.

165. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space Inches
65	8 & 12	8½	12	75	308	54	27	2	3	6	5	108 x 44
65	9 & 14	8½	12	75	308	54	27	2	3	6	5	109 x 44
65	8 & 12	10	12	75	427	54	27	2	3	8	7	112 x 46
65	9 & 14	10	12	75	427	54	27	2	3	8	7	113 x 46
65	10 & 16	10	12	75	427	54	27	2	3	8	7	115 x 46
65	9 & 14	12	12	75	615	54	27	2	3	10	8	114 x 49
65	10 & 16	12	12	75	615	54	27	2	3	10	8	115 x 49
65	12 & 18	12	12	75	615	54	27	2½	3½	10	8	117 x 49
65	10 & 16	14	12	75	840	54	27	2	3	12	10	118 x 53
65	12 & 18	14	12	75	840	54	27	2½	3½	12	10	120 x 53
65	14 & 20	14	12	75	840	54	27	2½	5	12	10	120 x 53
65	10 & 16	16	12	75	1090	54	27	2	3	14	12	128 x 55
65	12 & 18	16	12	75	1090	54	27	2½	3½	14	12	130 x 55
65	14 & 20	16	12	75	1090	54	27	2½	5	14	12	130 x 55
65	12 & 17	15	15	75	1210	68	27	2½	3½	12	10	156 x 64
65	14 & 20	15	15	75	1210	68	27	2½	5	12	10	156 x 64
65	12 & 17	17	15	75	1550	68	27	2½	3½	14	12	156 x 64
65	14 & 20	17	15	75	1550	68	27	2½	5	14	12	156 x 64

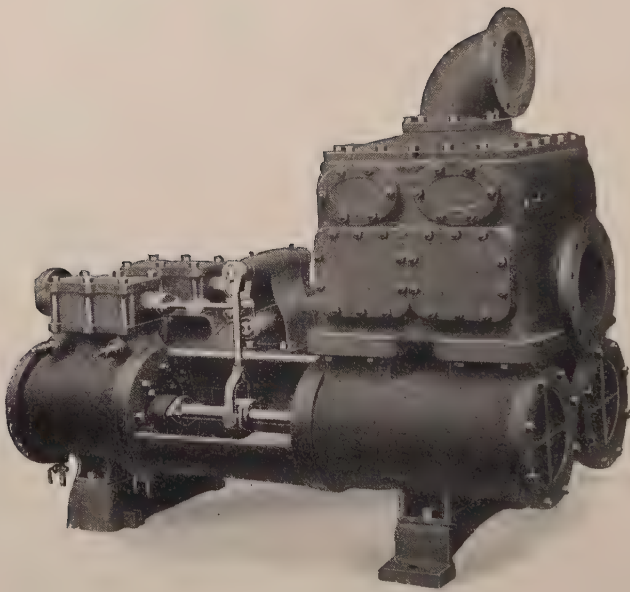


FIG. 66.

WORTHINGTON DUPLEX PACKED-PISTON PUMP

Rectangular Valve-box Pattern

FOR LOW-PRESSURE SERVICE

WORTHINGTON DUPLEX PACKED-PISTON PUMPS

Rectangular Valve-Box Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

166. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space Inches
66	12	15	12	50	960	54	27	2½	3	12	10	88 x 40
66	14	15	12	50	960	54	27	2½	3	12	10	86 x 40
66	16	15	12	50	960	54	27	2½	3	12	10	86 x 46
66	12	14	15	50	1060	68	27	2½	3	12	10	94 x 46
66	14	14	15	50	1060	68	27	2½	3	12	10	99 x 46
66	12	15	15	50	1210	68	27	2½	3	12	10	94 x 46
66	14	15	15	50	1210	68	27	2½	3	12	10	99 x 46
66	12	17	15	40	1550	68	27	2½	3	14	12	102 x 47
66	14	17	15	40	1550	68	27	2½	3	14	12	107 x 47
66	17	17	15	40	1550	68	27	2½	3½	14	12	107 x 52
66	14	19	15	40	1940	68	27	2½	3	16	14	103 x 49
66	17	19	15	40	1940	68	27	2½	3½	16	14	103 x 52
66	20	19	15	40	1940	68	27	4	5	16	14	103 x 49
66	14	22	15	40	2600	68	27	2½	3	16	14	109 x 55
66	17	22	15	40	2600	68	27	2½	3½	16	14	109 x 55
66	20	22	15	40	2600	68	27	4	5	16	14	109 x 55

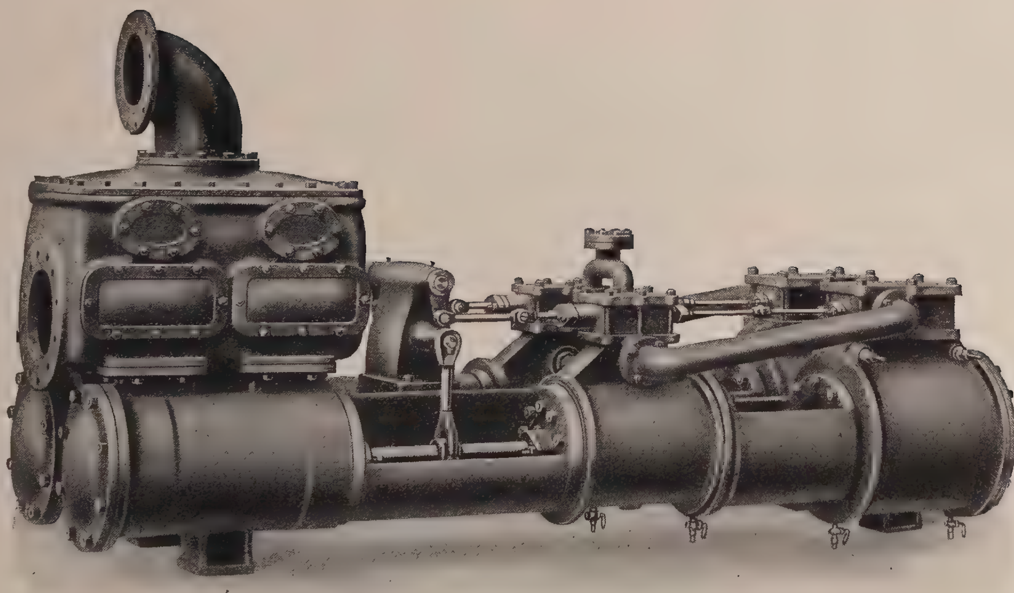


Fig. 67.

Worthington Compound Duplex Packed-Piston Pump

Rectangular Valve-box Pattern

For Low-pressure Service

WORTHINGTON COMPOUND DUPLEX PACKED-PISTON PUMPS

Rectangular Valve-Box Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

167. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Size of Pump in Inches: Diameter Liquid Pistons	Size of Pump in Inches: Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Capacity for Continuous Operation: Piston Speed Ft. per min.	Capacity for Continuous Operation: Rev. per min.	Pipe Sizes, Inches: Steam	Pipe Sizes, Inches: Exhaust	Pipe Sizes, Inches: Suction	Pipe Sizes, Inches: Discharge	Floor Space Inches
67	9 & 14	15	12	50	960	54	27	2	3	12	10	113 x 46
67	10 & 16	15	12	50	960	54	27	2	3	12	10	114 x 46
67	12 & 18	15	12	50	960	54	27	2½	3½	12	10	116 x 46
67	9 & 14	14	15	50	1060	68	27	2	3	12	10	137 x 48
67	12 & 17	14	15	50	1060	68	27	2½	3½	12	10	141 x 52
67	9 & 14	15	15	50	1210	68	27	2	3	12	10	137 x 48
67	12 & 17	15	15	50	1210	68	27	2½	3½	12	10	141 x 52
67	14 & 20	15	15	50	1210	68	27	2½	5	12	10	141 x 55
67	9 & 14	17	15	40	1550	68	27	2	3	14	12	145 x 49
67	12 & 17	17	15	40	1550	68	27	2½	3½	14	12	149 x 52
67	14 & 20	17	15	40	1550	68	27	2½	5	14	12	149 x 55
67	12 & 17	19	15	40	1940	68	27	2½	3½	16	14	145 x 52
67	14 & 20	19	15	40	1940	68	27	2½	5	16	14	145 x 55
67	12 & 17	22	15	40	2600	68	27	2½	3½	16	14	151 x 55
67	14 & 20	22	15	40	2600	68	27	2½	5	16	14	151 x 55

WORTHINGTON DUPLEX PACKED-PLUNGER PUMPS

170. Worthington packed-plunger pumps are designed for heavy, severe service where it is necessary to repack the moving parts of the liquid end without loss of time. They are suitable for boiler feeding and should always be used for pumping out mines, excavations, etc., or wherever the liquid pumped contains considerable quantities of sand or grit of any kind.

171. The plunger stuffing boxes are outside the liquid cylinders and excessive leakage is at once apparent. The arrangement of the plungers and stuffing boxes makes it possible to repack the plungers easily and quickly without opening up the liquid end and thereby keep the machine always at its highest pumping efficiency.

172. Worthington duplex packed-plunger pumps are available for the following capacities:

Gal. per min. Water and Similar Liquids	B. Hp. Boiler Feeding	Max. Disch. Pressure Lb. per sq. in.	Table No.
20– 80	180– 725	200	173
90– 330	870–3150	200–250	174
230– 855	2100–7900	200	175–176
340– 685	....	250	177
400–1550	....	200–250	178
400–5350	....	125–250	179–181
300–1260	....	75	182

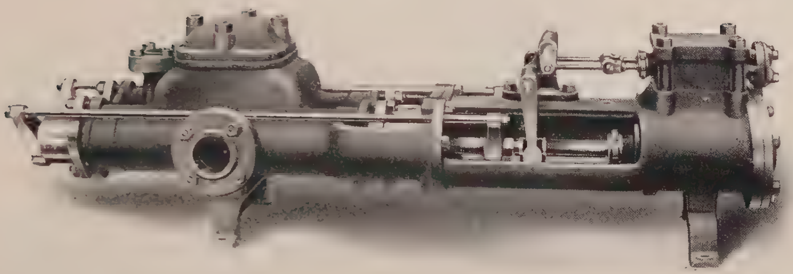


Fig. 68.

WORTHINGTON DUPLEX END-PACKED PLUNGER PUMPS

Ram Pattern. Valve-Plate Type

Maximum working pressure: Steam end—150 lb.
Liquid end—200 lb.

173. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation				Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinders	Diameter Plungers	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. of Boilers Pump Will Feed	Steam	Exhaust	Suction	Discharge	
68	4½	2¾	4	200	20	33	50	180	½	¾	2	1½	53 x 14
68	5¼	3½	5	200	36	38	45	325	¾	1¼	2½	1½	62 x 18
68	6	4	6	200	51	40	40	460	1	1¼	3	2	71 x 19
68	7½	5	6	200	80	40	40	725	1½	2	4	3	75 x 21

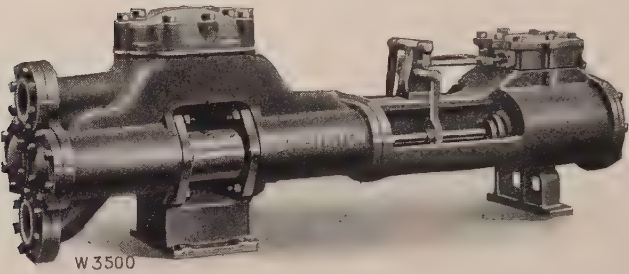


FIG. 69.

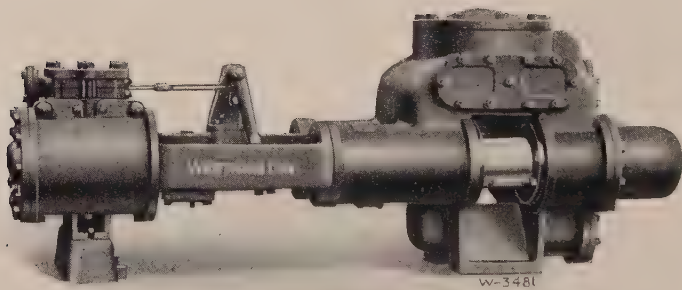


FIG. 70.

WORTHINGTON DUPLEX CENTER-PACKED PLUNGER PUMPS

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

174. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinder	Diameter Plungers	Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boilers Pump Will Feed	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space Inches
69	7½	4½	10	250	94	58	35	870	1½	2	4	3	83 x 22
69	9	5¼	10	250	126	58	35	1170	2	2½	4	3	86 x 23
69	10	6	10	250	165	58	35	1515	2	2½	5	4	88 x 26
70	12	7	10	200	225	58	35	2050	2	2½	6	5	95 x 31
70	12	8½	10	200	330	58	35	3150	2	2½	6	5	95 x 31
70	14	8½	10	200	330	58	35	3150	2½	3	6	5	101 x 33

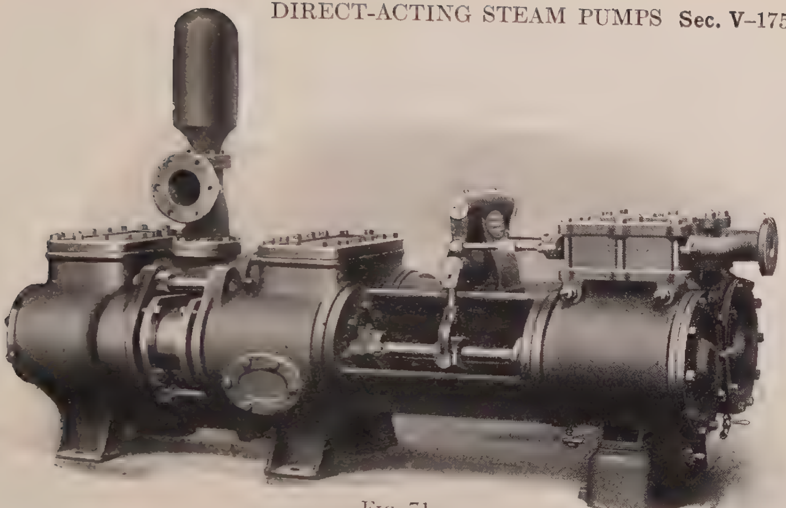


Fig. 71.

WORTHINGTON DUPLEX CENTER-PACKED PLUNGER PUMP

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

175. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good For	Capacity for Continuous Operation				Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinder	Diameter Plungers	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boilers Pump Will Feed	Steam	Exhaust	Suction	Discharge	
71	12	7	12	200	233	60	30	2100	2½	3	6	5	101 x 36
71	14	7	12	200	233	60	30	2100	2½	3	6	5	101 x 37
71	12	8½	12	200	344	60	30	3200	2½	3	6	5	111 x 36
71	14	8½	12	200	344	60	30	3200	2½	3	6	5	111 x 37
71	16	8½	12	200	344	60	30	3200	2½	3	6	5	112 x 46
71	18	8½	12	200	344	60	30	3200	3	3½	6	5	113 x 48
71	14	10¼	12	200	500	60	30	4700	2½	3	8	7	113 x 43
71	16	10¼	12	200	500	60	30	4700	2½	3	8	7	113 x 46
71	18	10¼	12	200	500	60	30	4700	3	3½	8	7	114 x 48
71	20	10¼	12	200	500	60	30	4700	4	5	8	7	115 x 50
71	16	12	12	200	685	60	30	6450	2½	3	10	8	113 x 46
71	18	12	12	200	685	60	30	6450	3	3½	10	8	114 x 48
71	20	12	12	200	685	60	30	6450	4	5	10	8	115 x 50
71	14	8½	15	200	428	75	30	4000	2½	3	6	5	115 x 43
71	17	8½	15	200	428	75	30	4000	2½	3½	6	5	115 x 47
71	17	10¼	15	200	625	75	30	5900	2½	3½	8	7	119 x 47
71	20	10¼	15	200	625	75	30	5900	4	5	8	7	121 x 50
71	17	12	15	200	855	75	30	7900	2½	3½	10	8	121 x 47
71	20	12	15	200	855	75	30	7900	4	5	10	8	122 x 50

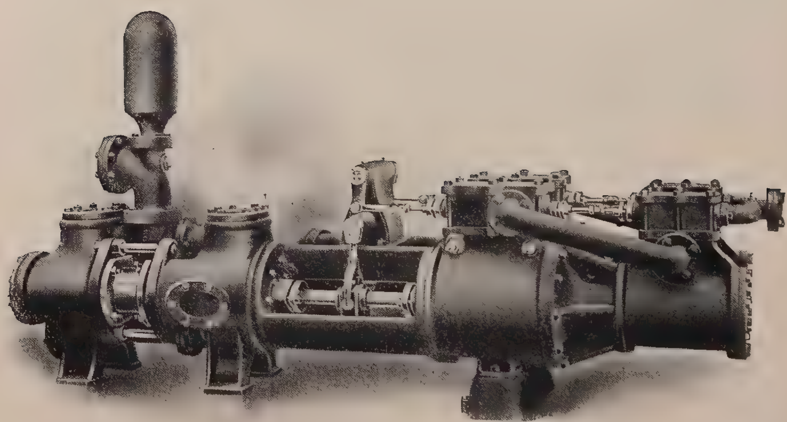


FIG. 72.

WORTHINGTON COMPOUND DUPLEX CENTER-PACKED PLUNGER PUMP

Scranton Pattern

WORTHINGTON COMPOUND DUPLEX CENTER-PACKED PLUNGER PUMPS

Scranton Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

176. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation				Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinders	Diameter Plungers	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boilers Pump Will Feed	Steam	Exhaust	Suction	Discharge	
72	8 & 12	7	12	200	233	60	30	2100	2	3	6	5	127 x 36
72	9 & 14	7	12	200	233	60	30	2100	2	3	6	5	128 x 37
72	8 & 12	8½	12	200	344	60	30	3200	2	3	6	5	128 x 37
72	9 & 14	8½	12	200	344	60	30	3200	2	3	6	5	130 x 37
72	10 & 16	8½	12	200	344	60	30	3200	2	3	6	5	132 x 46
72	12 & 18	8½	12	200	344	60	30	3200	2½	3½	6	5	132 x 48
72	9 & 14	10¼	12	200	500	60	30	4700	2	3	8	7	130 x 43
72	10 & 16	10¼	12	200	500	60	30	4700	2	3	8	7	133 x 46
72	12 & 18	10¼	12	200	500	60	30	4700	2½	3½	8	7	135 x 48
72	14 & 20	10¼	12	200	500	60	30	4700	2½	5	8	7	136 x 50
72	10 & 16	12	12	200	685	60	30	6450	2	3	10	8	134 x 46
72	12 & 18	12	12	200	685	60	30	6450	2½	3½	10	8	137 x 48
72	14 & 20	12	12	200	685	60	30	6450	2½	5	10	8	138 x 50
72	9 & 14	8½	15	200	428	75	30	4000	2	3	6	5	154 x 43
72	12 & 17	8½	15	200	428	75	30	4000	2½	3½	6	5	156 x 47
72	14 & 20	8½	15	200	428	75	30	4000	2½	5	6	5	156 x 50
72	12 & 17	10¼	15	200	625	75	30	5900	2½	3½	8	7	160 x 47
72	14 & 20	10¼	15	200	625	75	30	5900	2½	5	8	7	160 x 50
72	12 & 17	12	15	200	855	75	30	7900	2½	3½	10	8	162 x 47
72	14 & 20	12	15	200	855	75	30	7900	2½	5	10	8	162 x 50

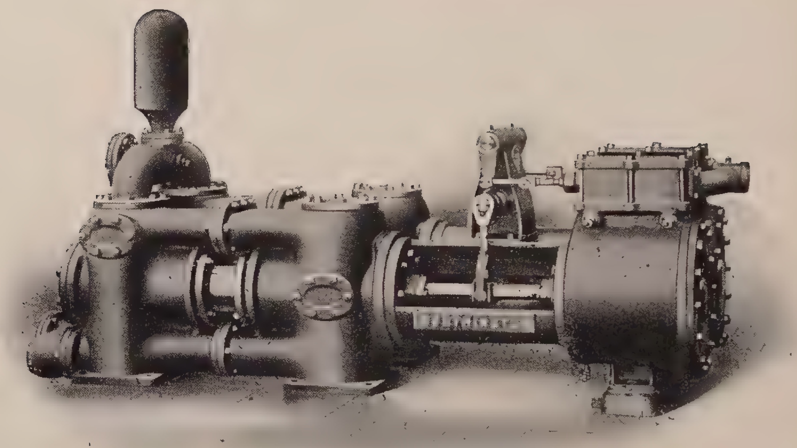


Fig. 73.

Worthington Simple Duplex Center-Packed Plunger Pump

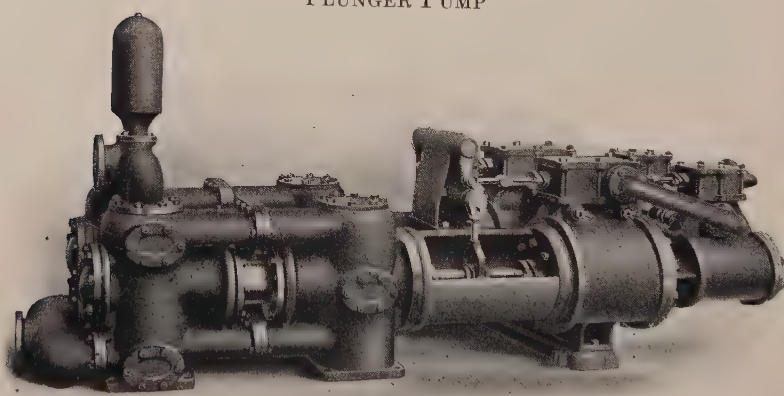


Fig. 74.

Worthington Compound Duplex Center-Packed Plunger Pump

Scranton Pattern

WORTHINGTON DUPLEX CENTER-PACKED PLUNGER PUMPS

Simple and Compound

Scranton Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end—250 lb.

These pumps may also be furnished with triple-expansion steam ends. See list of sizes in table, par. 95.

177. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation			Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinders	Diameter Plungers	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
73	14	8½	12	250	344	60	30	2½	3	8	6	122 x 38
73	16	8½	12	250	344	60	30	2½	3	8	6	124 x 46
73	18	8½	12	250	344	60	30	3	3½	8	6	124 x 48
73	16	10¼	12	250	500	60	30	2½	3	10	8	135 x 46
73	18	10¼	12	250	500	60	30	3	3½	10	8	135 x 48
73	20	10¼	12	250	500	60	30	4	5	10	8	135 x 50
73	16	12	12	250	685	60	30	2½	3	12	10	136 x 50
73	18	12	12	250	685	60	30	3	3½	12	10	136 x 50
73	20	12	12	250	685	60	30	4	5	12	10	136 x 50
73	17	8½	15	250	428	75	30	2½	3½	8	6	126 x 49
73	20	8½	15	250	428	75	30	4	5	8	6	127 x 50
73	17	10¼	15	250	625	75	30	2½	3½	10	8	139 x 49
73	20	10¼	15	250	625	75	30	4	5	10	8	140 x 50
73	20	12	15	250	855	75	30	4	5	12	10	142 x 52
74	8 & 12	8½	12	250	344	60	30	2	3	8	6	146 x 38
74	9 & 14	8½	12	250	344	60	30	2	3	8	6	147 x 38
74	10 & 16	8½	12	250	344	60	30	2	3	8	6	148 x 46
74	12 & 18	8½	12	250	344	60	30	2½	3½	8	6	149 x 48
74	9 & 14	10¼	12	250	500	60	30	2	3	10	8	158 x 45
74	10 & 16	10¼	12	250	500	60	30	2	3	10	8	159 x 46
74	12 & 18	10¼	12	250	500	60	30	2½	3½	10	8	160 x 48
74	14 & 20	10¼	12	250	500	60	30	2½	5	10	8	161 x 50
74	10 & 16	12	12	250	685	60	30	2	3	12	10	160 x 50
74	12 & 18	12	12	250	685	60	30	2½	3½	12	10	161 x 50
74	14 & 20	12	12	250	685	60	30	2½	5	12	10	162 x 50

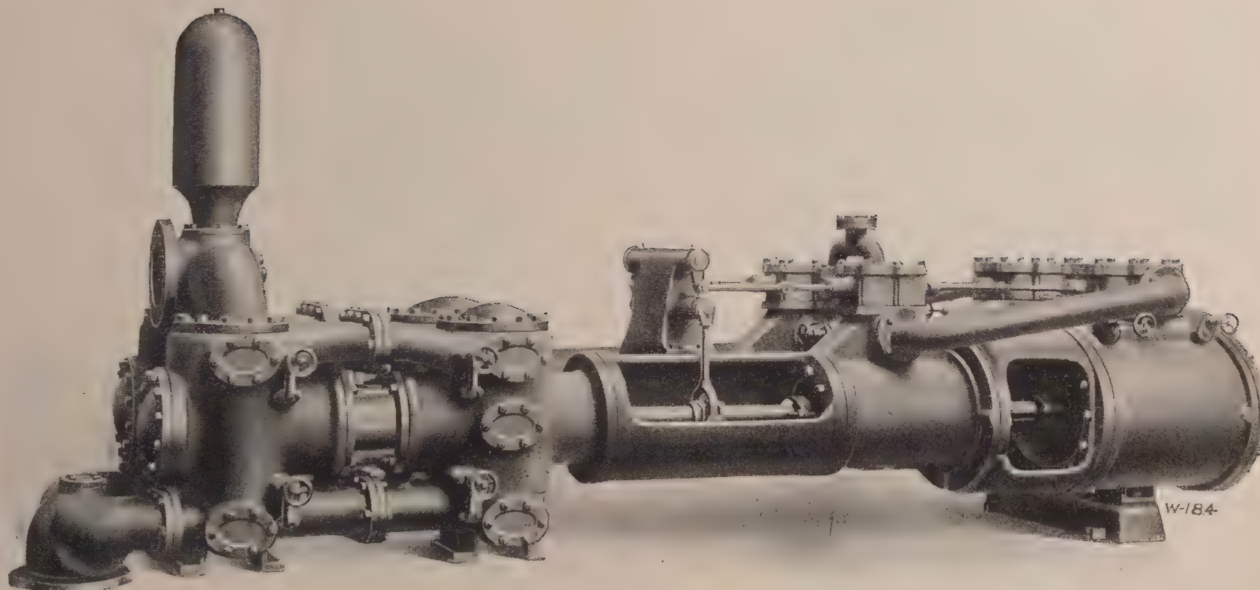


Fig. 75.

Worthington Compound Duplex Center-Packed
Plunger Pump

Scranton Pattern

WORTHINGTON COMPOUND DUPLEX CENTER-PACKED PLUNGER PUMPS

Scranton Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

These pumps may also be furnished with triple-expansion steam ends. See list of sizes in table 95.

178. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Diameter Plungers	Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space Feet and Inches
75	14 & 20	8½	15	250	428	75	30	2½	5	8	6	14- 0 x 4- 2
75	14 & 20	10¼	15	250	625	75	30	2½	5	10	8	14-10 x 4- 2
75	14 & 20	12	15	250	855	75	30	2½	5	12	10	15- 0 x 4- 3
75	10 & 16	8½	18	250	515	90	30	2½	3½	8	6	16- 4 x 3-10
75	12 & 18	8½	18	250	515	90	30	2½	4	8	6	16-10 x 3-10
75	14 & 22	8½	18	250	515	90	30	2½	6	8	6	18- 0 x 4- 6
75	12 & 18	10	18	250	712	90	30	2½	4	10	8	17- 6 x 3-10
75	14 & 22	10	18	250	712	90	30	2½	6	10	8	18- 6 x 4- 6
75	14 & 22	12	18	250	1025	90	30	2½	6	12	10	18- 6 x 4- 6
75	16 & 25	12	18	250	1025	90	30	3	6	12	10	18- 6 x 4- 6
75	16 & 25	14	18	200	1400	90	30	3	6	12	10	18- 9 x 5- 0
75	10 & 16	8½	24	250	570	100	25	2½	3½	8	6	17- 8 x 3-10
75	12 & 18	8½	24	250	570	100	25	2½	4	8	6	17-11 x 4- 4
75	14 & 22	8½	24	250	570	100	25	2½	6	8	6	18- 0 x 4- 9
75	12 & 18	10	24	250	790	100	25	2½	4	10	8	18- 9 x 4- 4
75	14 & 22	10	24	250	790	100	25	2½	6	10	8	18-10 x 4- 9
75	14 & 22	12	24	250	1140	100	25	2½	6	12	10	19- 1 x 4- 9
75	16 & 25	12	24	250	1140	100	25	3	6	12	10	19- 3 x 5- 0
75	16 & 25	14	24	200	1550	100	25	3	6	12	10	20- 0 x 5 -6

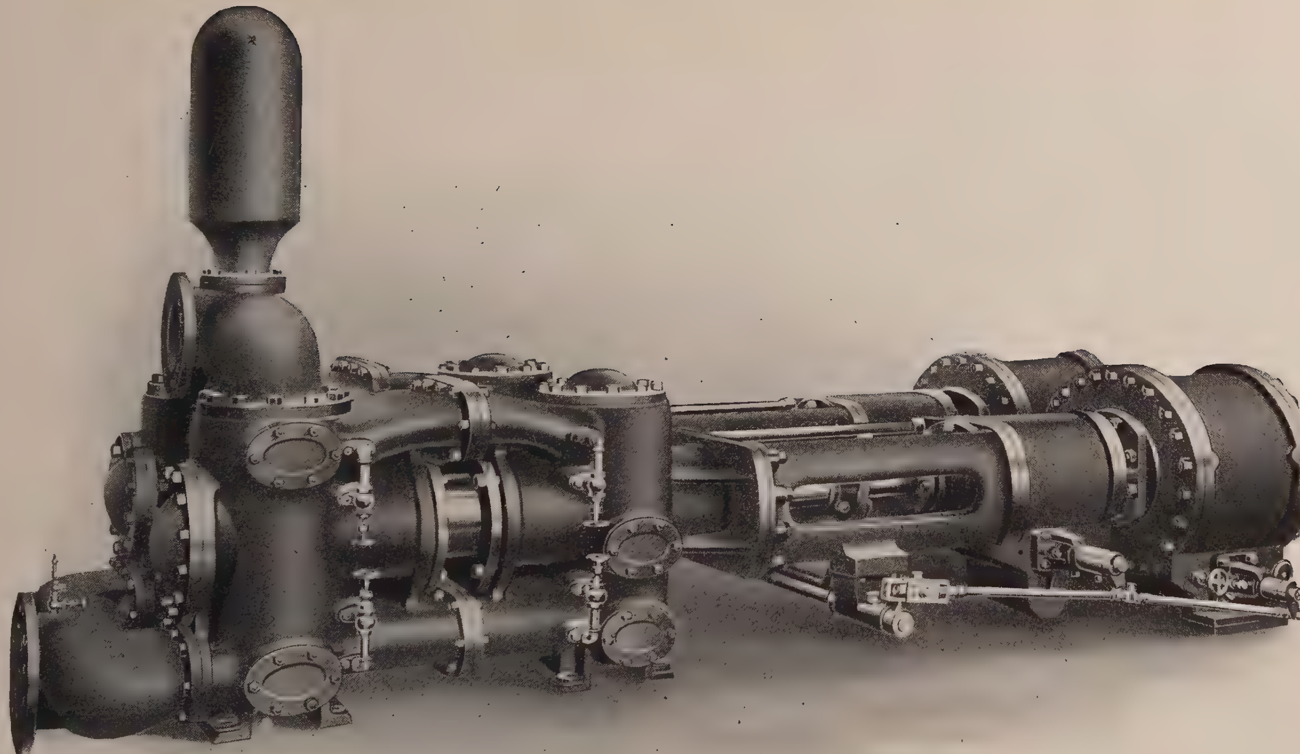


Fig. 76.

Worthington Compound Duplex Center-Packed Plunger Pump

Steam Valves, Rotative Type

WORTHINGTON COMPOUND DUPLEX CENTER-PACKED PLUNGER PUMPS

With Rotative Steam Valves

Scranton Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end—See table

179. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Size of Pump in Inches: Diameter Plungers	Size of Pump in Inches: Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Capacity for Continuous Operation: Piston Speed Ft. per min.	Capacity for Continuous Operation: Rev. per min.	Pipe Sizes, Inches: Steam	Pipe Sizes, Inches: Exhaust	Pipe Sizes, Inches: Suction	Pipe Sizes, Inches: Discharge	Floor Space, Feet and Inches
76	14 & 20	8½	15	250	428	75	30	3	5	8	6	12– 9 x 5–7
76	14 & 20	10¼	15	250	625	75	30	3	5	10	8	13– 2 x 5–7
76	16 & 25	10¼	15	250	625	75	30	4	7	10	8	14– 7 x 7–0
76	14 & 20	12	15	250	855	75	30	3	5	12	10	13– 4 x 5–7
76	16 & 25	12	15	250	855	75	30	4	7	12	10	14– 9 x 7–0
76	12 & 18	8½	18	250	515	90	30	2½	4	8	6	13– 4 x 6–0
76	14 & 22	8½	18	250	515	90	30	3	7	8	6	13– 6 x 6–2
76	12 & 18	10¼	18	250	745	90	30	2½	4	10	8	13– 9 x 6–0
76	14 & 22	10¼	18	250	745	90	30	3	7	10	8	13–11 x 6–2
76	16 & 25	12	18	250	1025	90	30	4	7	12	10	14– 9 x 7–0
76	18 & 29	12	18	250	1025	90	30	4	8	12	10	14–10 x 7–0
76	14 & 22	14	18	200	1400	90	30	3	7	12	10	15– 7 x 6–2
76	16 & 25	14	18	200	1400	90	30	4	7	12	10	16– 3 x 7–0
76	18 & 29	14	18	200	1400	90	30	4	8	12	10	16– 4 x 7–0
76	16 & 25	16	18	200	1820	90	30	4	7	16	14	16– 9 x 7–0
76	18 & 29	16	18	200	1820	90	30	4	8	16	14	16–10 x 7–0
76	18 & 29	18	18	200	2300	90	30	4	8	18	16	18– 0 x 7–0
76	16 & 25	12	24	250	1140	100	25	4	7	12	10	18– 0 x 7–0
76	19 & 30	12	24	250	1140	100	25	4	8	12	10	18– 6 x 8–4

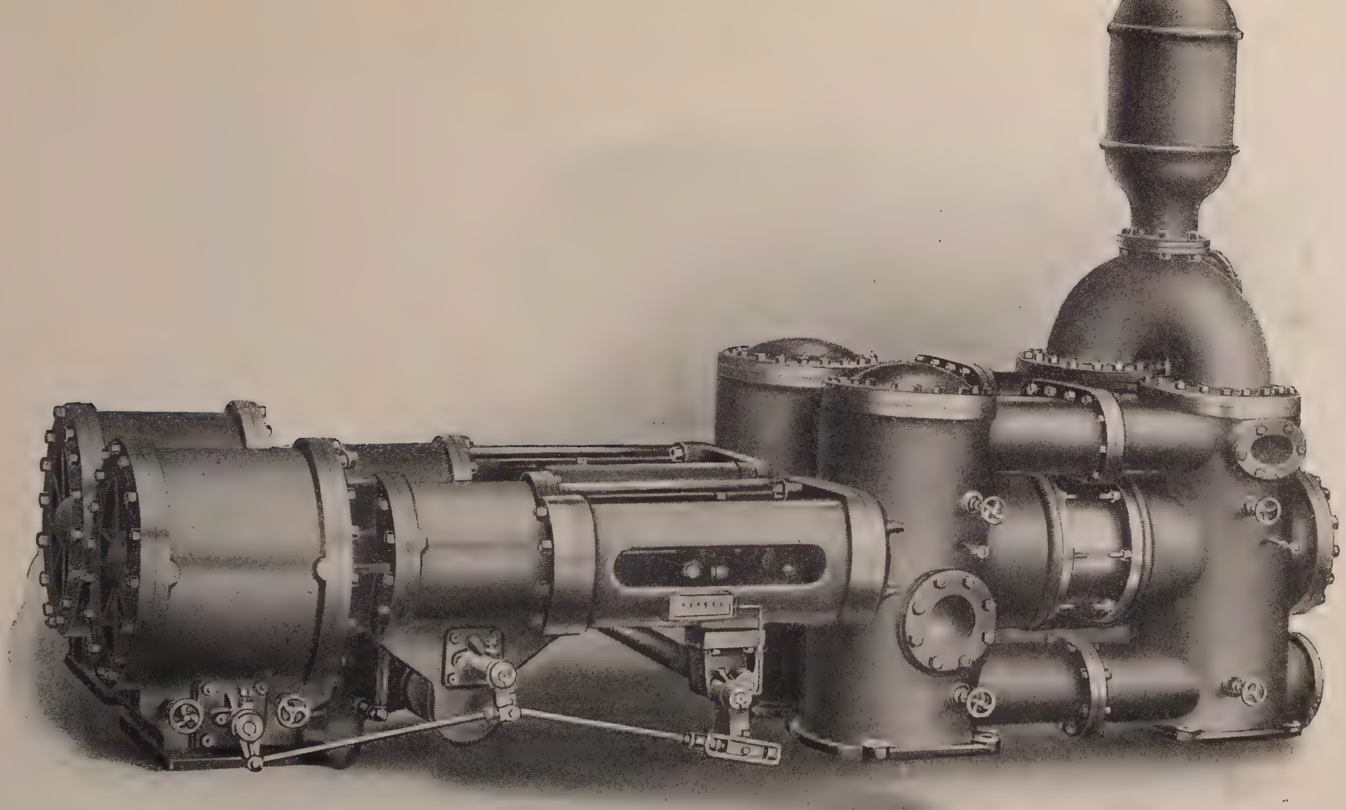


FIG. 77.
WORTHINGTON COMPOUND DUPLEX CENTER-PACKED PLUNGER PUMP
Scranton Pattern
STEAM VALVES, ROTATIVE TYPE

WORTHINGTON DUPLEX CENTER-PACKED PLUNGER PUMPS

With Rotative Steam Valves

Scranton Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

These pumps can also be furnished with triple expansion-steam ends. See list of sizes in table 95.

180. TABLE OF SIZES AND DATA (Concluded on following page)

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Size of Pump in Inches: Diameter Plungers	Size of Pump in Inches: Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Capacity for Continuous Operation: Piston Speed Ft. per min.	Capacity for Continuous Operation: Rev. per min.	Pipe Sizes, Inches: Steam	Pipe Sizes, Inches: Exhaust	Pipe Sizes, Inches: Suction	Pipe Sizes, Inches: Discharge	Floor Space Feet and Inches
77	16 & 25	14	24	200	1550	100	25	4	7	16	14	19– 9 x 6–11
77	19 & 30	14	24	200	1550	100	25	4	8	16	14	20– 3 x 8– 4
77	21 & 34	14	24	200	1550	100	25	4	8	16	14	20– 5 x 8–10
77	16 & 25	16	24	200	2025	100	25	4	7	18	16	21– 4 x 6–11
77	19 & 30	16	24	200	2025	100	25	4	8	18	16	21–10 x 8– 4
77	21 & 34	16	24	200	2025	100	25	4	8	18	16	22– 0 x 8–10
77	23 & 38	16	24	200	2025	100	25	4	8	18	16	22– 1 x 9– 9
77	16 & 25	18	24	200	2655	100	25	4	7	20	18	21– 9 x 6–11
77	19 & 30	18	24	200	2655	100	25	4	8	20	18	22– 3 x 8– 4
77	21 & 34	18	24	200	2655	100	25	4	8	20	18	22– 5 x 8–10
77	23 & 38	18	24	200	2655	100	25	4	8	20	18	22– 6 x 9– 9
77	25 & 42	18	24	200	2655	100	25	4	8	20	18	22– 6 x 10– 3
77	16 & 25	20	24	200	3165	100	25	4	7	20	18	21– 9 x 6–11
77	19 & 30	20	24	200	3165	100	25	4	8	20	18	22– 3 x 8– 4
77	21 & 34	20	24	200	3165	100	25	4	8	20	18	22– 5 x 8–10
77	23 & 38	20	24	200	3165	100	25	4	8	20	18	22– 6 x 9– 9
77	25 & 42	20	24	200	3165	100	25	4	8	20	18	22– 6 x 10– 3

WORTHINGTON COMPOUND DUPLEX CENTER-PACKED PLUNGER PUMPS

With Rotative Steam Valves

Scranton Pattern

181. TABLE OF SIZES AND DATA—Concluded

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation			Pipe Sizes, Inches				Floor Space Feet and Inches
	Diameter Steam Cylinders	Diameter Plungers	Stroke		Gal. per Min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
77	16 & 25	22	24	175	3830	100	25	4	7	24	20	23– 0 x 6–11
77	19 & 30	22	24	175	3830	100	25	4	8	24	20	23– 6 x 8– 4
77	21 & 34	22	24	175	3830	100	25	4	8	24	20	23– 8 x 8–10
77	23 & 38	22	24	175	3830	100	25	4	8	24	20	23– 9 x 10– 0
77	25 & 42	22	24	175	3830	100	25	4	8	24	20	23– 9 x 10– 3
77	19 & 30	24	24	150	4555	100	25	4	8	24	20	23– 7 x 8– 4
77	21 & 34	24	24	150	4555	100	25	4	8	24	20	23– 9 x 8–10
77	23 & 38	24	24	150	4555	100	25	4	8	24	20	23–10 x 10– 0
77	25 & 42	24	24	150	4555	100	25	4	8	24	20	23–10 x 10– 3
77	19 & 30	25	24	150	5050	100	25	4	8	24	20	23– 7 x 8– 4
77	21 & 34	25	24	150	5050	100	25	4	8	24	20	23– 9 x 8–10
77	23 & 38	25	24	150	5050	100	25	4	8	24	20	23–10 x 10– 0
77	25 & 42	25	24	150	5050	100	25	4	8	24	20	23–10 x 10– 3
77	19 & 30	26	24	125	5350	100	25	4	8	30	24	23– 7 x 8– 4
77	21 & 34	26	24	125	5350	100	25	4	8	30	24	23– 9 x 9– 1
77	23 & 38	26	24	125	5350	100	25	4	8	30	24	23–10 x 10– 0
77	25 & 42	26	24	125	5350	100	25	4	8	30	24	23–10 x 10– 3

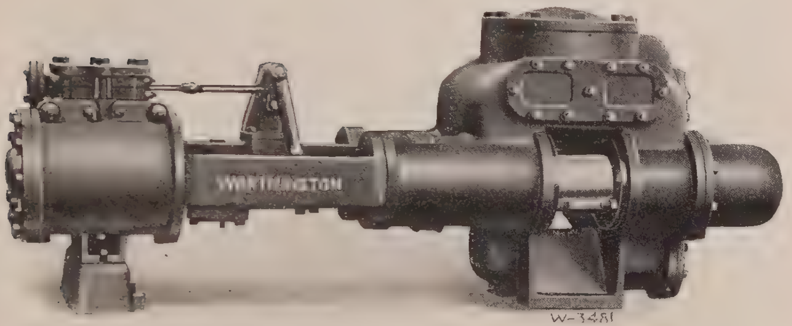


Fig. 78.

WORTHINGTON DUPLEX CENTER-PACKED PLUNGER PUMPS

Maximum working pressure: Steam end—150 lb.
Liquid end— 75 lb.

182. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation			Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinders	Diameter Plungers	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
78	8	8½	12	75	308	54	27	1½	2	6	5	101 x 30
78	10	8½	12	75	308	54	27	2	2½	6	5	103 x 30
78	10	10¼	12	75	448	54	27	2	2½	8	7	103 x 36
78	12	10¼	12	75	448	54	27	2½	3	8	7	109 x 36
78	10	12	12	75	615	54	27	2	2½	12	10	106 x 39
78	12	12	12	75	615	54	27	2½	3	12	10	111 x 39
78	16	14	12	75	840	54	27	2½	3	14	12	120 x 48
78	10	10¼	18	75	670	81	27	2	2½	8	7	130 x 36
78	12	10¼	18	75	670	81	27	2	2½	8	7	130 x 36
78	10	12	18	75	922	81	27	2	2½	12	10	130 x 39
78	12	12	18	75	922	81	27	2½	3	12	10	130 x 39
78	14	12	18	75	922	81	27	2½	3	12	10	130 x 39
78	16	14	18	75	1260	81	27	2½	3½	14	12	143 x 48

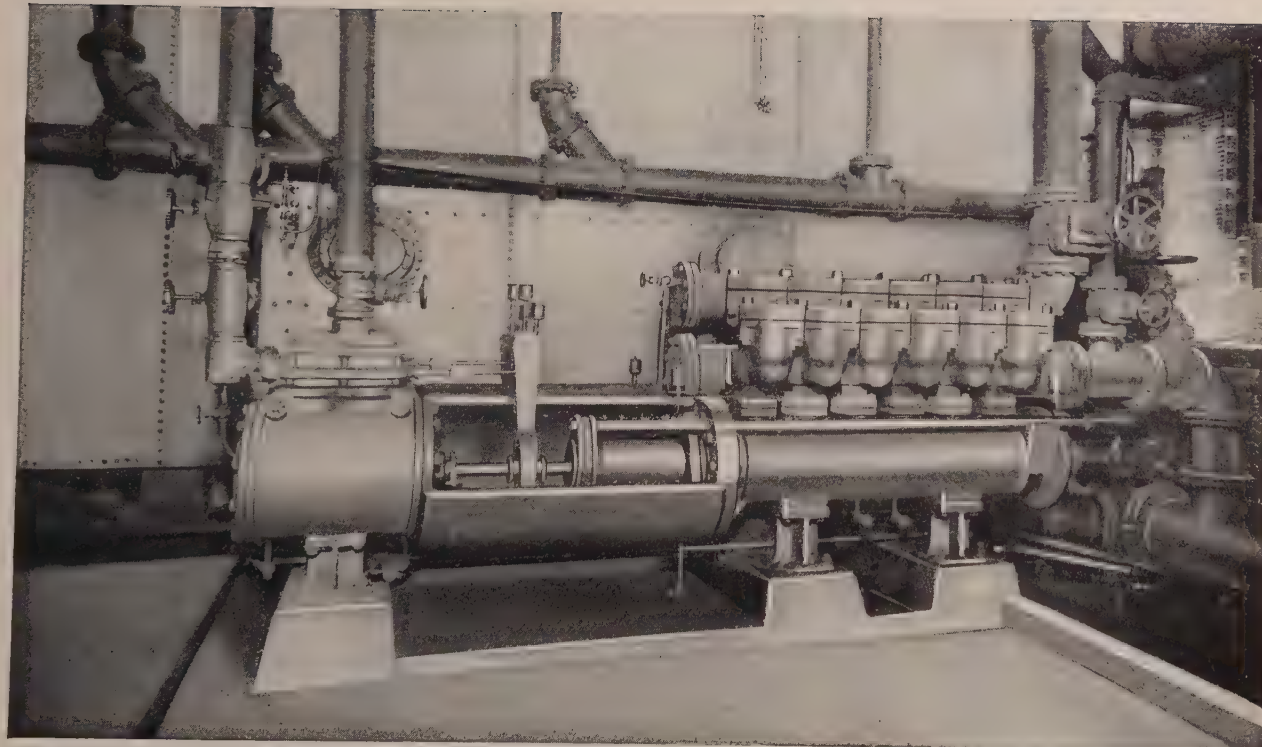


Fig. 78a.
Worthington, Duplex Outside End packed-plunger, Pot-valve Boiler-feed Pumps in a Sugar Refinery

WORTHINGTON DUPLEX PRESSURE PUMPS

300 to 12,000 lb. per sq. in. pressure

183. Worthington duplex pressure pumps have liquid ends of the pot-valve type for discharge pressures up to 2000 lb. per sq. in. and of the forged-steel type for pressures of 2000 lb. per sq. in. and upwards, both types having outside-packed plungers.

184. The Worthington 300 lb. per sq. in. pot-pattern pressure pumps of both the end-packed plunger and center-packed plunger patterns were designed to meet the demand for a high-grade pump for boiler feed service in modern high-pressure power plants. These pumps are unexcelled for this exacting service.

185. For general hydraulic service, where the pressure exceeds 300 lb. per sq. in., special lines of pot-pattern and forged-steel liquid ends have been developed in the sizes listed in the following tables:

Gal. per Min. Water and Similar Liquids	B. Hp. Boiler Feeding	Max. Disch. Pressure Lb. per sq. in.	Table No.
17– 213	240–3100	300	186
274– 547	4000–7900	300	187–188
1.3– 30	....	600– 2000	189
21– 257	....	600– 2000	191
27– 324	....	600– 2000	192
100– 428	....	500– 2000	193
1– 40	....	2000–12,000	190
350–1400	....	400– 600	194

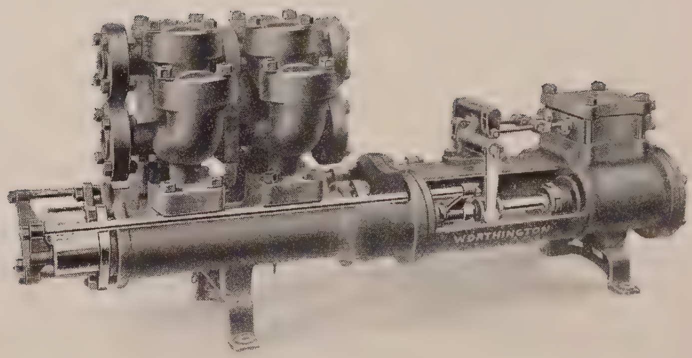


FIG. 79.

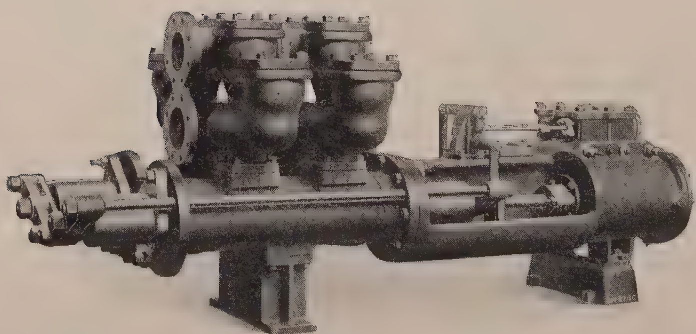


FIG. 80.

WORTHINGTON DUPLEX END-PACKED PLUNGER PUMPS

Valve-Pot Type

WORTHINGTON DUPLEX END-PACKED PLUNGER PUMPS

Valve-Pot Type

Maximum working pressure: Steam end—175 lb.
Liquid end—300 lb.

186. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Capacity for Continuous Operation							Pipe Sizes, Inches				Floor Space Inches
				As a Boiler Feed Pump				As a Pressure Pump							
	Diameter of Steam Cylinders	Diameter of Plungers	Stroke	Gal. per min.	Hp. Boilers Pump Will Feed	Piston Speed Ft. per min.	Rev. per min.	Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
79	5¼	3	5	17	240	24	29	24	34	41	¾	1¼	2½	2	59 x 19
79	6	3½	6	25	360	26	26	34	36	36	1	1¼	2½	2	67 x 19
79	7½	4½	6	41	600	26	26	57	36	36	1½	2	4	3	70 x 25
79	7½	4½	10	60	870	38	23	82	52	31	1½	2	4	3	99 x 25
79	9	5	10	74	1050	38	23	101	52	31	2	2½	4	3	99 x 25
79	10	6	12	108	1575	39	20	151	54	27	2	2½	5	4	108 x 37
80	12	7	12	148	2100	39	20	205	54	27	2½	3	6	5	108 x 43
80	12	7½	15	213	3100	49	20	296	63	27	2½	3	6	5	141 x 51

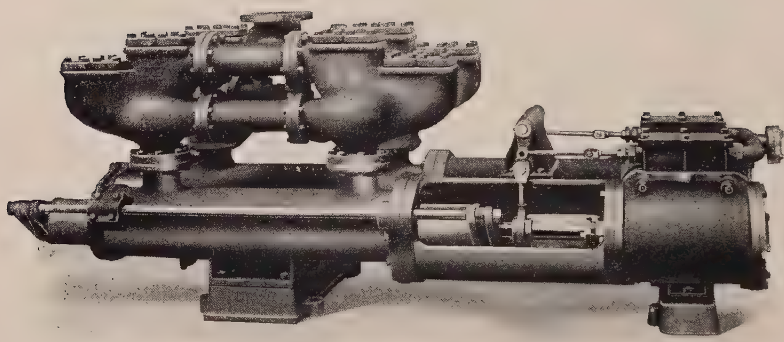


Fig. 81.

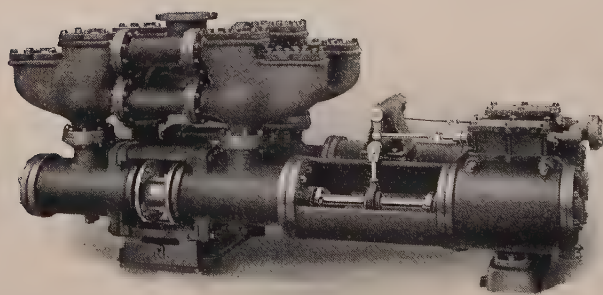


Fig. 82.

WORTHINGTON DUPLEX PACKED PLUNGER PUMPS

END AND CENTER-PACKED PATTERNS

Valve-Pot Type

WORTHINGTON DUPLEX PACKED-PLUNGER PUMPS

End and Center-Packed Patterns

Valve-Pot Type

Maximum working pressure: Steam end— 175 lb.
Liquid end—300 lb.

187. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Capacity for Continuous Operation							Pipe Sizes, Inches				Floor Space Inches
				As a Boiler Feed Pump				As a Pressure Pump							
	Diameter of Steam Cylinders	Diameter Plungers	Stroke	Gal. per min.	Hp. Boilers Pump will feed	Piston Speed Ft. per min.	Rev. per min.	Gal. per min.	Piston Speed ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
END-PACKED															
81	14	8½	15	274	4000	49	20	380	68	27	2½	3	8	7	145 x 45
81	17	10	15	380	5500	49	20	528	68	27	2½	3½	8	7	150 x 48
81	20	12	15	547	7900	49	20	760	68	27	4	5	10	8	159 x 71
CENTER-PACKED															
82	14	8½	15	274	4000	49	20	380	68	27	2½	3	8	7	127 x 42
82	17	10	15	380	5500	49	20	528	68	27	2½	3½	8	7	135 x 43
82	20	12	15	547	7900	49	20	760	68	27	4	5	10	8	146 x 71

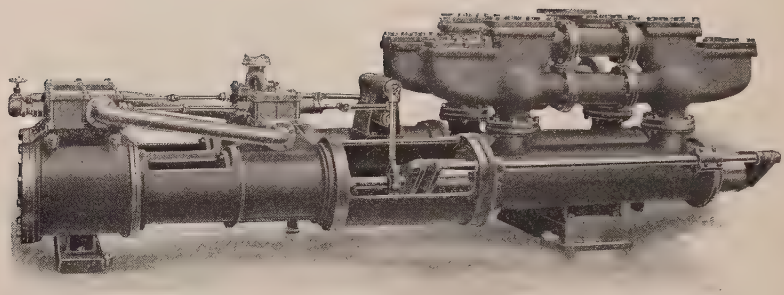


Fig. 83.

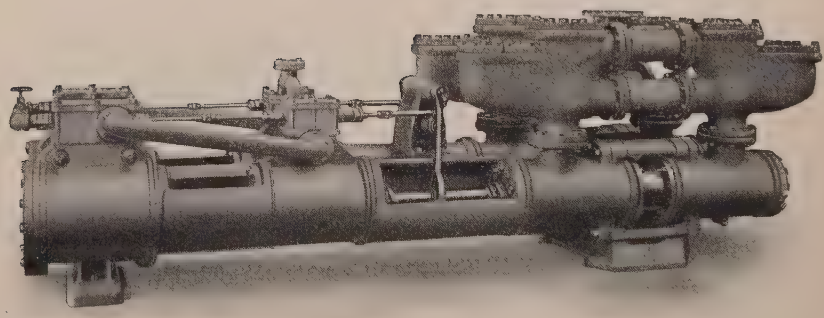


Fig. 84.

Worthington Compound Duplex Packed-Plunger Pumps

End and Center-Packed Patterns

Valve-Pot Type

WORTHINGTON COMPOUND DUPLEX PACKED-PLUNGER PUMP

End and Center-Packed Patterns

Valve-Pot Type

Maximum working pressure: Steam end—175 lb.
Liquid end—300 lb.

188. TABLE OF SIZES AND DATA

	Size of Pump in Inches			Capacity for Continuous Operation							Pipe Sizes, Inches				
				As a Boiler Feed Pump				As a Pressure Pump							Floor Space Feet and Inches
Fig No.	Diameter of Steam Cylinders	Diameter of Plungers	Stroke	Gal. per min.	Hp. of Boilers Pump Will Feed	Piston Speed Ft. per min.	Rev. per min.	Gal. per min.	Piston Speed Ft. per min.	Rev. per. min.	Steam	Exhaust	Suction	Discharge	
							END-PACKED								
83	9 & 14	8½	15	274	4000	49	20	380	68	27	2	3	8	7	13–3x3– 8
83	12 & 17	8½	15	274	4000	49	20	380	68	27	2½	3½	8	7	15–3x3–11
83	12 & 17	10	15	380	5500	49	20	528	68	27	2½	3½	8	7	15–6x3–11
83	14 & 20	10	15	380	5500	49	20	528	68	27	2½	5	8	7	16–1x3–11
83	14 & 20	12	15	547	7900	49	20	760	68	27	2½	5	10	8	16–9x5–11
							CENTER-PACKED								
84	9 & 14	8½	15	274	4000	49	20	380	68	27	2	3	8	7	13–3x3– 9
84	12 & 17	8½	15	274	4000	49	20	380	68	27	2½	3½	8	7	14–0x3–11
84	12 & 17	10	15	380	5500	49	20	528	68	27	2½	3½	8	7	15–0x4– 3
84	14 & 20	10	15	380	5500	49	20	528	68	27	2½	5	8	7	15–6x4– 3
84	14 & 20	12	15	547	7900	49	20	760	68	27	2½	5	10	8	18–3x5–11

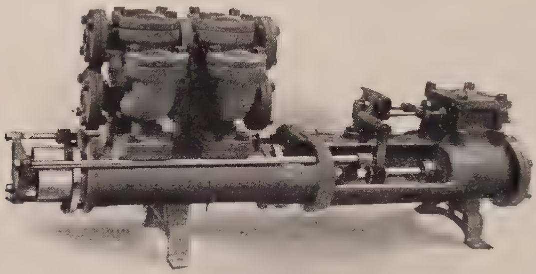


Fig. 85.

WORTHINGTON DUPLEX END-PACKED PLUNGER PUMPS

Valve-Pot Type

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

189. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Maximum Discharge Pressure Good for	Capacity for Continuous Operation			Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinders	Diameter of Plungers	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
85	4½	¾	4	2000	1.3	30	45	½	¾	¾	¾	49 x 29
85	4½	1½	4	650	5.0	30	45	½	¾	1½	1¼	49 x 29
85	5¼	¾	5	2000	1.5	35	42	¾	1¼	¾	¾	61 x 28
85	5¼	1¾	5	650	8.0	35	42	¾	1¼	1½	1¼	61 x 28
85	6	¾	6	2000	1.6	36	36	1	1¼	¾	¾	67 x 28
85	6	1¾	6	800	8.7	36	36	1	1¼	1½	1¼	67 x 28
85	7½	1¾	6	800	8.7	36	36	1½	2	1½	1¼	63 x 28
85	9	1¾	6	800	8.7	36	36	1½	2	1½	1¼	71 x 28
85	6	2½	6	800	18.	36	36	1	1¼	2	1½	67 x 28
85	7½	2½	6	800	18.	36	36	1½	2	2	1½	68 x 28
85	9	2½	6	800	18.	36	36	1½	2	2	1½	71 x 28
85	6	3¼	6	600	30.	36	36	1	1¼	2	1½	67 x 28
85	7½	3¼	6	600	30.	36	36	1½	2	2	1½	68 x 28
85	9	3¼	6	600	30.	36	36	1½	2	2	1½	71 x 28

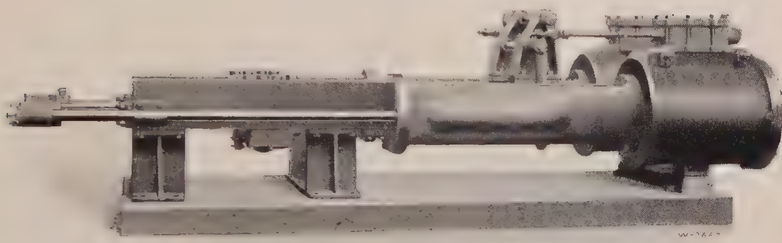


FIG. 86.

WORTHINGTON DUPLEX END-PACKED PLUNGER PUMPS

FORGED-STEEL LIQUID END

Maximum working pressure: Steam end—150 lb.
Liquid end—12,000 lb.

190. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter of Steam Cylinders	Size of Pump in Inches: Diameter of Plungers	Size of Pump in Inches: Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Capacity for Continuous Operation: Piston Speed Ft. per min.	Capacity for Continuous Operation: Rev. per min.	Pipe Sizes, Inches: Steam	Pipe Sizes, Inches: Exhaust	Pipe Sizes, Inches: Suction	Pipe Sizes, Inches: Discharge	Floor Space Inches
86	6	1	6	2000 lb. to 12,000 lb.	1.75	30	30	1	1¼	To Suit Conditions	To Suit Conditions	83 x 16
86	7½	1	6		1.75	30	30	1½	2			83 x 16
86	8	1	12		2.9	50	25	1½	2			106 x 28
86	10	1¼	12		6.3	50	25	2½	2			108 x 34
86	12	1¼	12		6.3	50	25	2½	3			110 x 36
86	12	1½	12		8.5	50	25	2½	3			110 x 36
86	14	1¾	12		12.	50	25	2½	3			127 x 43
86	16	1¾	12		12.	50	25	2½	3			127 x 45
86	17	2¼	15		24.	60	24	2½	3½			148 x 48
86	20	2¼	15		24.	60	24	4	5			150 x 52
86	18	2½	18		35.	70	23	3	4			163 x 52
86	20	2½	18		35.	70	23	4	5			165 x 52
86	20	2¾	18		42.	70	23	4	5			165 x 52
86	20	3	18		50.	70	23	4	5			165 x 52
86	25	3½	18		68.	70	23	5	6			180 x 66
86	25	2¾	24		40.	80	20	5	6			212 x 72

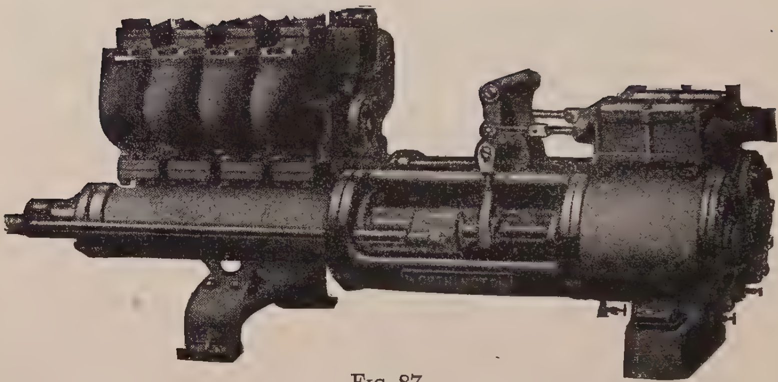


FIG. 87

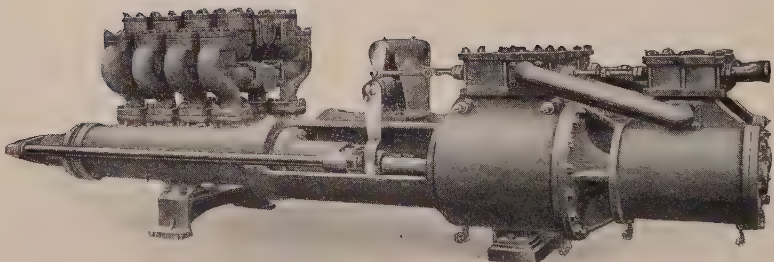


FIG. 88.

WORTHINGTON DUPLEX END-PACKED PLUNGER PUMPS
SIMPLE AND COMPOUND
Valve-Pot Type

WORTHINGTON DUPLEX END-PACKED PLUNGER PUMPS

Simple and Compound

Valve-Pot Type (Figs. 87 and 88)

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

191. TABLE No. 1. LIQUID ENDS

Size of liquid end in inches: Diameter of Plungers	Stroke	Number of Pots on Liquid End	Maximum Discharge Pressure Good for: Cast Iron	Cast Steel	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Diameter of Suction Pipe	Diameter of Discharge Pipe
2¼	12	4	800	2000	21	54	27	2½	2
2½	12	4	800	2000	26	54	27	2½	2
2¾	12	4	800	2000	32	54	27	2½	2
3	12	4	800	2000	38	54	27	2½	2
3¼	12	4	800	2000	45	54	27	2½	2
3½	12	4	800	2000	52	54	27	2½	2
3¾	12	4	800	1950	60	54	27	2½	2
4	12	4	600	1200	68	54	27	4	3
4¼	12	4	600	1200	78	54	27	4	3
4½	12	4	600	1200	86	54	27	4	3
4¾	12	8	600	1450	96	54	27	4	3
5	12	8	600	1225	107	54	27	4	3
5¼	12	8	600	1150	117	54	27	4	3

Size of Liquid end in inches: Diameter of Plungers	Stroke	Number of Pots on Liquid End	Maximum Discharge Pressure Good for: Cast Iron	Cast Steel	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Diameter of Suction Pipe	Diameter of Discharge Pipe
5½	12	8	600	1075	129	54	27	4	3
5¾	12	8	600	1025	141	54	27	5	4
6	12	8	600	1000	153	54	27	5	4
6¼	12	8	600	1000	167	54	27	5	4
6½	12	8	600	1000	180	54	27	5	4
6¾	12	12	600	1200	195	54	27	6	5
7	12	12	600	1200	210	54	27	6	5
7¼	12	12	600	1125	225	54	27	6	5
7½	12	12	600	1050	241	54	27	6	5
7¾	12	12	600	1000	257	54	27	6	5

When selecting a pressure pump, the liquid end is taken from this table and combined with the proper size steam end taken from Table No. 2. The size of the steam end to be used depends upon the discharge pressure of the pump and the steam pressure available at the pump throttle. Rules for selecting the proper size steam end are given in Par. 70.

TABLE No. 2. STEAM ENDS

Size of Simple Steam End in Inches: Diameter of Steam Cylinders	Stroke	Diameter of Steam Pipe in Inches	Diameter of Exhaust Pipe in Inches	Size of Compound Steam End in Inches: Diameter of Steam Cylinders	Stroke	Diameter of Steam Pipe in Inches	Diameter of Exhaust Pipe in Inches
10	12	2	2½	....	....	....	....
12	12	2½	3	8 & 12	12	2	3
14	12	2½	3	9 & 14	12	2	3
16	12	2½	3	10 & 16	12	2	3
18	12	3	3½	12 & 18	12	2½	3½
20	12	4	5	14 & 20	12	2½	5

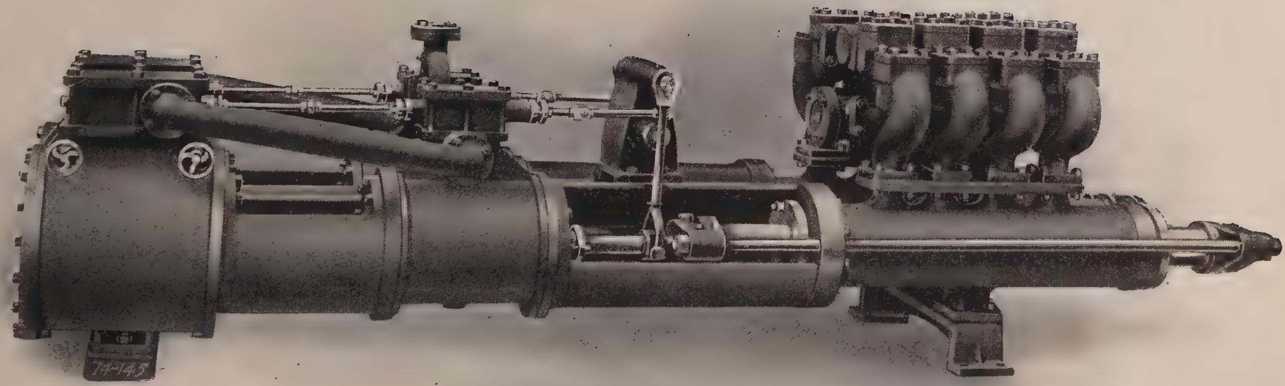


Fig. 89.

Worthington Compound Duplex End-Packed Plunger Pump

Valve-Pot Type

WORTHINGTON DUPLEX END-PACKED PLUNGER PUMPS

Simple and Compound

Valve-Pot Type (Figs. 89 and 90.)

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

192. TABLE No. 1. LIQUID ENDS

Size of liquid end in inches			Maximum Discharge Pressure Good For		Capacity for Continuous Operation				
Diameter of Plungers	Stroke	Number of Pots on Liquid End	Cast Iron	Cast Steel	Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Diameter of Suction Pipe	Diameter of Discharge Pipe
2¼	15	4	800	2000	27	68	27	2½	2
2½	15	4	800	2000	34	68	27	2½	2
2¾	15	4	800	2000	41	68	27	2½	2
3	15	4	800	2000	49	68	27	2½	2
3¼	15	4	800	2000	58	68	27	2½	2
3½	15	4	800	2000	66	68	27	2½	2
3¾	15	4	800	1950	76	68	27	2½	2
4	15	4	600	1200	87	68	27	4	3
4¼	15	4	600	1200	97	68	27	4	3
4½	15	4	600	1200	109	68	27	4	3
4¾	15	8	600	1450	122	68	27	4	3
5	15	8	600	1225	135	68	27	4	3
5¼	15	8	600	1150	148	68	27	4	3
5½	15	8	600	1075	162	68	27	4	3
5¾	15	8	600	1025	178	68	27	5	4
6	15	8	600	1000	195	68	27	5	4
6¼	15	8	600	1000	211	68	27	5	4
6½	15	8	600	1000	228	68	27	5	4
6¾	15	12	600	1200	244	68	27	6	5
7	15	12	600	1200	266	68	27	6	5
7¼	15	12	600	1125	284	68	27	6	5
7½	15	12	600	1050	302	68	27	6	5
7¾	15	12	600	1000	324	68	27	6	5

When selecting a pressure pump, the liquid end is taken from this table and combined with the proper size steam end taken from table No. 2. The size of the steam end to be used depends upon the discharge pressure of the pump and the steam pressure available at the pump throttle. Rules for selecting the proper size steam end are given in Par. 79.

TABLE No. 2. STEAM ENDS

Size of Simple Steam End in Inches		Diameter of Steam Pipe in Inches	Diameter of Exhaust Pipe in Inches	Size of Compound Steam End in Inches		Diameter of Steam Pipe in Inches	Diameter of Exhaust Pipe in Inches
Diameter of Steam Cylinders	Stroke			Diameter of Steam Cylinders	Stroke		
10	15	2	2½	9 & 14	15	2	3
12	15	2½	3	12 & 17	15	2½	3½
14	15	2½	3	14 & 20	15	2½	5
17	15	2½	3½	....	....	....	....
20	15	4	5	....	....	....	....

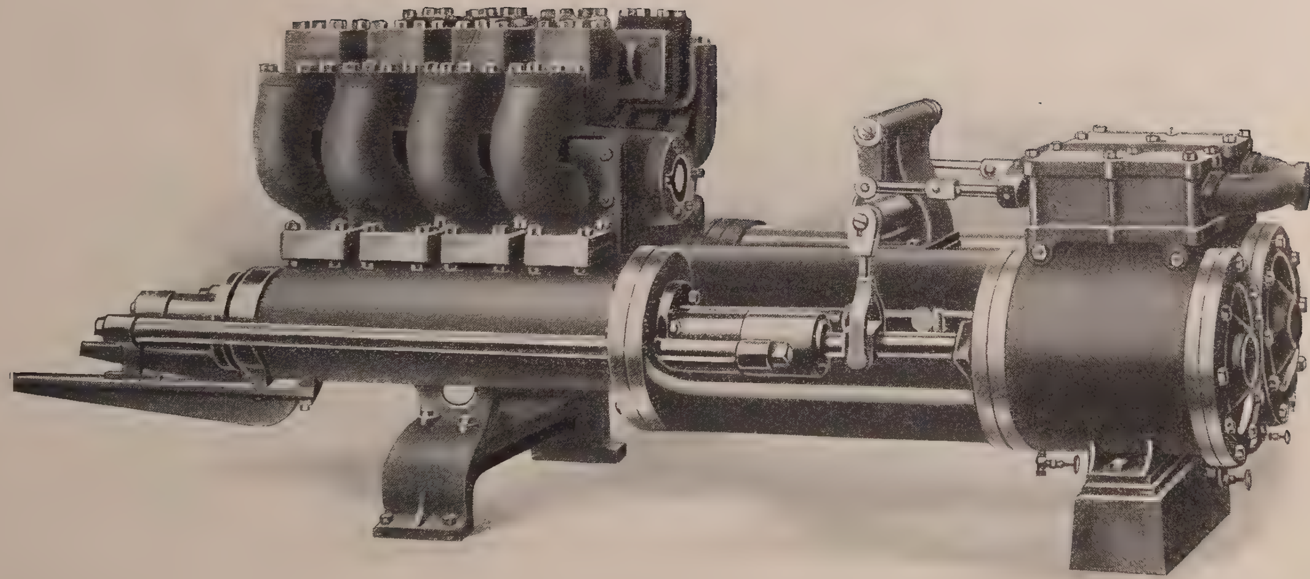


Fig. 90.

Worthington Duplex End-packed Plunger Pump
Valve-Pot Type

WORTHINGTON DUPLEX END-PACKED PLUNGER PUMPS

Valve-Pot Type (Fig. 90)

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

193. TABLE No. 1. LIQUID ENDS

Size of Liquid End in Inches			Maximum Discharge Pressure Good for		Capacity for Continuous Operation					Size of Liquid End in Inches			Maximum Discharge Pressure Good for		Capacity for Continuous Operation				
Diameter of Plungers	Stroke	Number of Pots on Liquid End	Cast Iron	Cast Steel	Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Diameter of Suction Pipe	Diameter of Discharge Pipe	Diameter of Plungers	Stroke	Number of Pots on Liquid End	Cast Iron	Cast Steel	Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Diameter of Suction Pipe	Diameter of Discharge Pipe
4	18	8	800	2000	101	80	27	4	3	4	24	8	800	2000	114	90	23	4	3
4½	18	8	800	2000	129	80	27	4	3	4½	24	8	800	2000	145	90	23	4	3
5	18	8	700	1750	159	80	27	4	3	5	24	8	700	1750	178	90	23	4	3
5½	18	8	600	1200	192	80	27	4	3	5½	24	8	600	1200	215	90	23	4	3
6	18	8	600	1200	228	80	27	5	4	6	24	8	600	1200	255	90	23	5	4
6½	18	12	500	1200	268	80	27	5	4	6½	24	12	500	1200	300	90	23	5	4
7	18	12	500	1200	310	80	27	6	5	7	24	12	500	1200	350	90	23	6	5
7½	18	12	500	1000	355	80	27	6	5	7½	24	12	500	1000	400	90	23	6	5
7¾	18	12	500	1000	380	80	27	6	5	7¾	24	12	500	1000	420	90	23	6	5

TABLE No. 2. STEAM ENDS

Size of Steam End in Inches				Size of Steam End in Inches			
Diameter of Steam Cylinders	Stroke	Diameter of Steam Pipe in Inches	Diameter of Exhaust Pipe in Inches	Diameter of Steam Cylinders	Stroke	Diameter of Steam Pipe in Inches	Diameter of Exhaust Pipe in Inches
14	18	2½	3	14	24	2½	3½
16	18	2½	3½	16	24	2½	3½
18	18	3	4	18	24	3	4
20	18	4	5	20	24	4	5
22	18	5	6	22	24	5	6
25	18	5	6	25	24	5	6

When selecting a pressure pump, the liquid end is taken from Table 1 and combined with the proper size steam end. The steam end may be taken from Table 2 on this page, or if a compound or triple expansion steam end is wanted use table 95. The size of steam end to be used depends upon the discharge pressure of the pump and the steam pressure available at the pump throttle. Rules for selecting the proper size steam end are given in par. 79.

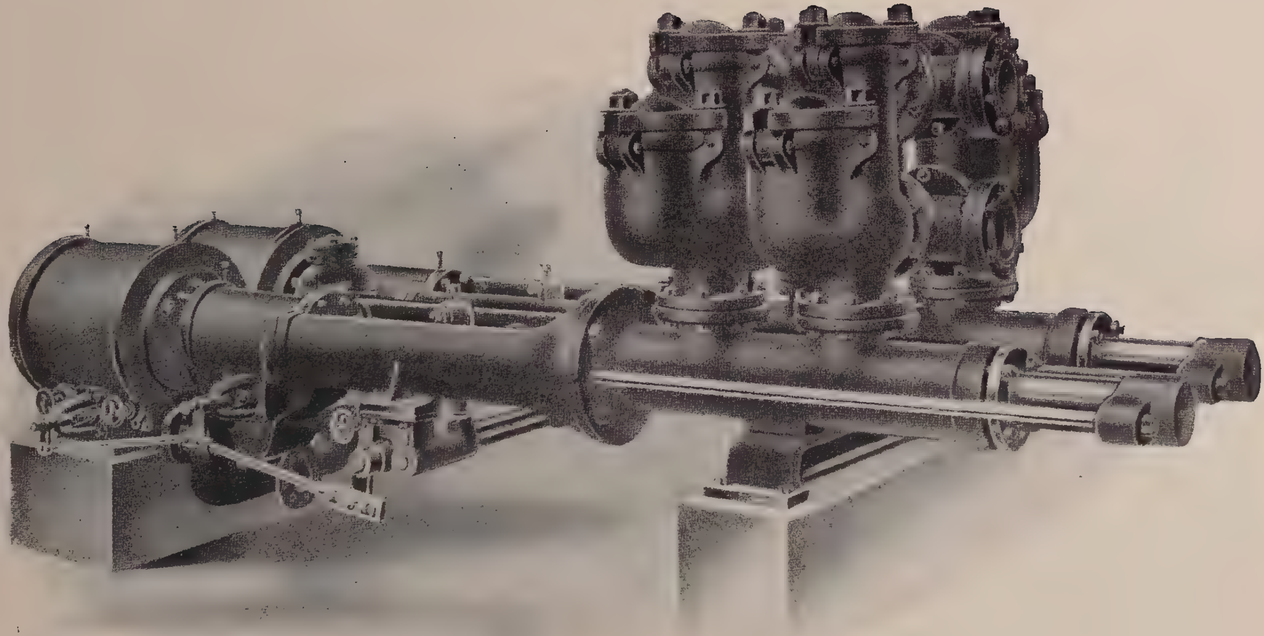


Fig. 91
Worthington Compound Duplex End-Packed Plunger Pumps
Lehigh Pattern
Rotative Steam Valves

WORTHINGTON COMPOUND DUPLEX END-PACKED PLUNGER PUMPS

Rotative Steam Valves

Lehigh Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

194. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter of Steam Cylinders	Diameter of Plungers	Stroke	No. Pots on Liquid End	Maximum Discharge Pressure Good for: Cast Iron	Semi-Steel	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space Feet and Inches
91	12 & 18	7½	18	4	400	600	356	80	27	2¼	4	To suit conditions	To suit conditions	17–0 x 7–0
91	14 & 20	7½	18	4	400	600	356	80	27	3	5			17–0 x 7–0
91	14 & 22	10	18	8	400	600	632	80	27	3	7			17–0 x 7–0
91	16 & 25	10	18	8	400	600	632	80	27	4	7			17–6 x 7–0
91	18 & 29	10	18	8	400	600	632	80	27	4	8			19–0 x 7–6
91	18 & 29	12	18	8	400	600	912	80	27	4	8			19–0 x 7–6
91	16 & 25	10	24	8	400	600	711	90	23	4	7			25–0 x 7–6
91	19 & 30	12	24	8	400	600	1026	90	23	4	8			25–0 x 8–0
91	21 & 34	12	24	8	400	600	1026	90	23	4	8			25–0 x 8–0
91	25 & 42	12	24	8	400	600	1026	90	23	4	8			25–0 x 8–0
91	21 & 34	14	24	12	400	600	1395	90	23	4	8			25–0 x 8–6
91	25 & 42	14	24	12	400	600	1395	90	23	4	8			25–0 x 8–6

These pumps can be furnished with either simple, compound or triple-expansion duplex steam ends when conditions require.

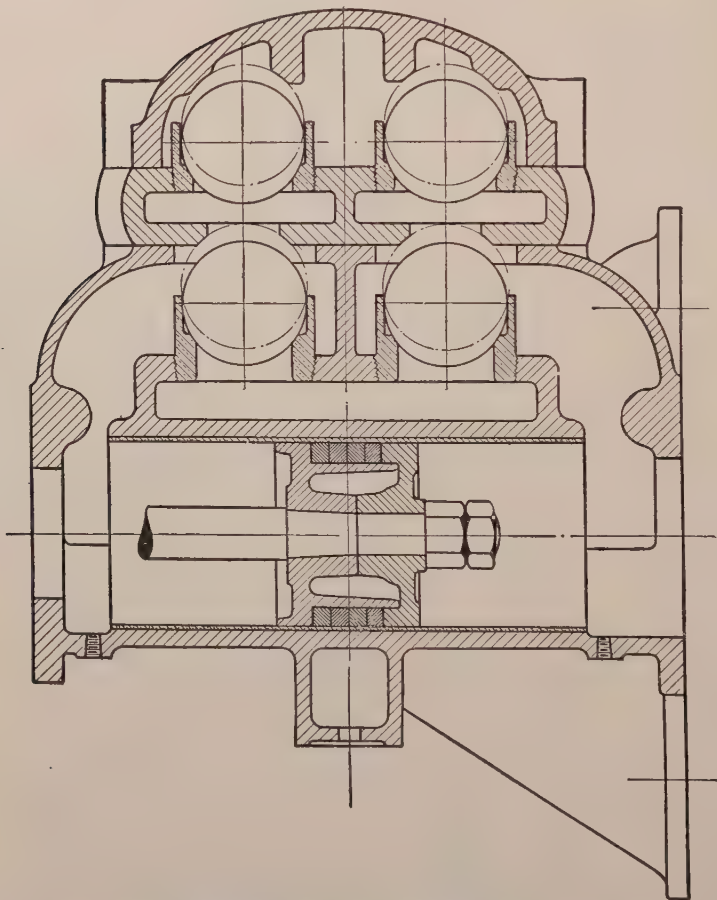


Fig. 92. Section through liquid end. Worthington Valve-plate Type Piston-pattern Pumps with ball valves.

195. For pumping tar, molasses and other thick viscous, liquids, we recommend the valve-plate pattern submerged piston pump with ball valves, as shown in Fig. 92.

196. This liquid end combines all the features of the valve-plate type submerged piston pump described in par. 41, with the ball-valve service, which is necessary for pumping thick liquids.

197. The ball valves can be of rubber, bronze or iron, depending upon the liquid to be handled.

WORTHINGTON DUPLEX PACKED PISTON PUMPS

Valve-Plate Type with Ball Valves (Fig. 92)

199. TABLE OF SIZES AND DATA

For Discharge Pressures up to 75 lb. per sq. in.

Size of Pump in inches			Capacity for Continuous Operation. Thick Liquids			Pipe Sizes, Inches				
Diameter of Steam Cylinders	Diameter of Liquid Pistons	Stroke	Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	Floor Space, Inches
4½	3¾	4	21	19	28	½	¾	2½	1½	35 x 13
5¼	4¾	5	38	21	25	¾	1¼	3	2	39 x 16
6	5¾	6	55	22	22	1	1¼	4	3	47 x 17
7½	7½	6	94	22	22	1½	2	6	5	49 x 22

200. TABLE OF SIZES AND DATA

For Discharge Pressures up to 200 lb. per sq. in.

Size of Pump in Inches			Capacity for Continuous Operation, Thick Liquids			Pipe Sizes, Inches				
Diameter of Steam Cylinders	Diameter of Liquid Pistons	Stroke	Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Delivery	Floor Space, Inches
4½	2¾	4	11	19	28	½	¾	2	1½	33 x 13
5¼	3½	5	20	21	25	¾	1¼	2½	1½	38 x 16
6	4	6	28	22	22	1	1¼	3	2	44 x 17
7½	5	6	43	22	22	1½	2	4	3	45 x 22
7½	4½	10	53	33	20	1½	2	4	3	59 x 22
9	5¼	10	72	33	20	2	2½	4	3	61 x 22
10	6	10	94	33	20	2	2½	5	4	62 x 36

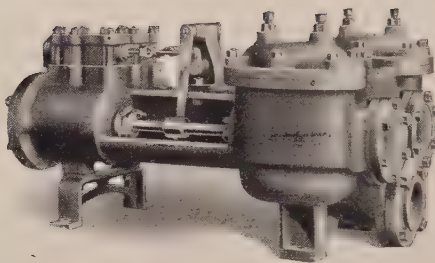


FIG. 93.

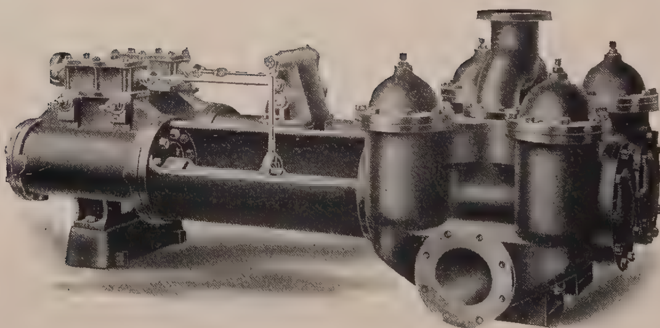


FIG. 94.

WORTHINGTON DUPLEX PACKED-PISTON PUMPS

Ball-Valve Pattern

WORTHINGTON DUPLEX PACKED-PISTON PUMP

Ball-Valve Pattern

For tar, molasses, and similar viscous liquids

Maximum working pressure: Steam end—150 lb.
Liquid end—see table

201. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinder	Diameter Liquid Pistons	Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Lb. (42 Gal.) per Hour	Piston Speed Ft. per min.	Rev. per min.	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space Inches
93	4½	2¾	4	125	11	16	19	28	½	¾	2½	2	34 x 23
93	4½	3¾	4	125	21	30	19	28	½	¾	2½	2	34 x 23
93	5½	3½	5	125	20	29	21	25	¾	1¼	3	2½	39 x 23
93	5¼	4¾	5	125	38	54	21	25	¾	1¼	4	3	40 x 27
93	6	4	6	125	28	40	22	22	1	1¼	3	2½	44 x 23
93	6	5¾	6	125	55	79	21	25	1	1¼	4	3	44 x 27
93	7½	5	6	125	43	62	22	22	1½	2	4	3	51 x 30
93	7½	4½	10	125	53	76	33	20	1½	2	4	3	60 x 30
93	9	5¼	10	125	72	103	33	20	2	2½	4	3	62 x 30
93	10	6	12	125	100	143	35	17.5	2	2½	5	4	68 x 30
94	7½	7½	18	75	215	307	48	16	1½	2	6	5	99 x 48
94	10	7½	18	75	215	307	48	16	2	2½	6	5	100 x 48
94	12	7½	18	75	215	307	48	16	2½	3	6	5	101 x 48
94	9	8½	18	75	275	395	48	16	2	2½	8	6	101 x 53
94	10	8½	18	75	275	395	48	16	2	2½	8	6	101 x 53
94	12	8½	18	75	275	395	48	16	2½	3	8	6	102 x 57
94	10	10	18	75	378	540	48	16	2	2½	8	6	102 x 57
94	12	10	18	75	378	540	48	16	2½	3	8	6	103 x 57
94	12	12	18	75	550	786	48	16	2½	3	10	8	110 x 72
94	14	12	18	75	550	786	48	16	2½	3	10	8	110 x 72
94	12	13	24	75	750	1071	56	14	2½	3	10	8	125 x 72
94	14	13	24	75	750	1071	56	14	2½	3	10	8	125 x 72

WORTHINGTON DUPLEX UNDERWRITER FIRE PUMPS

202. Worthington Underwriter Fire Pumps have water passages, valve areas, suction and discharge nozzles much larger than in ordinary pumps, to enable a greater amount of water to be discharged without water-hammer. The steam and exhaust ports and nozzles are designed so as to give unrestricted passage to the steam. Pumps are "rust-proof," and will start instantly after standing unused for a long time. The piston rods and valve rods are made from bronze. All stuffing boxes and glands are bronze lined. Plungers and plunger sleeves are of bronze, but the metals are of different compositions, to prevent cutting. This composition of metals, which has been a subject of much experiment and study, we have made up with great success; the parts work perfectly upon each other, without friction or impairment.

203. Each pump has the following **National Standard fittings,** included in the price and regularly furnished:

Capacity plate on discharge air chamber.
Stroke gage, graduated on each end.
Vacuum, or suction, air chamber.
Steam gage, 5-in. diameter.
Water gage, 5-in. diameter.
Relief-valve of large capacity.
Relief-valve discharge cone.
Set of brass priming pipes and special priming valves.
Required number of 2½-in. hose valves.
One sight-feed cylinder lubricator.
One hand oil pump.
All in strict accord with the "Underwriters" specifications.

204. On account of larger passageways, bronze parts and the special attachments mentioned, the Underwriter Pump necessarily costs more than an ordinary fire pump; but the **cost per gallon of water discharged** is less, since the Underwriter Pump can deliver a greater quantity of water in the same length of time. It is also much heavier and stronger, of superior workmanship, and better protected from rust and accident, than the ordinary fire pump.

205. Hose valves are threaded only when specially ordered, and at a small additional price. If we are to thread these valves, a sample thread must be supplied by the purchaser, as there is no established standard, hose threads varying widely in different localities.

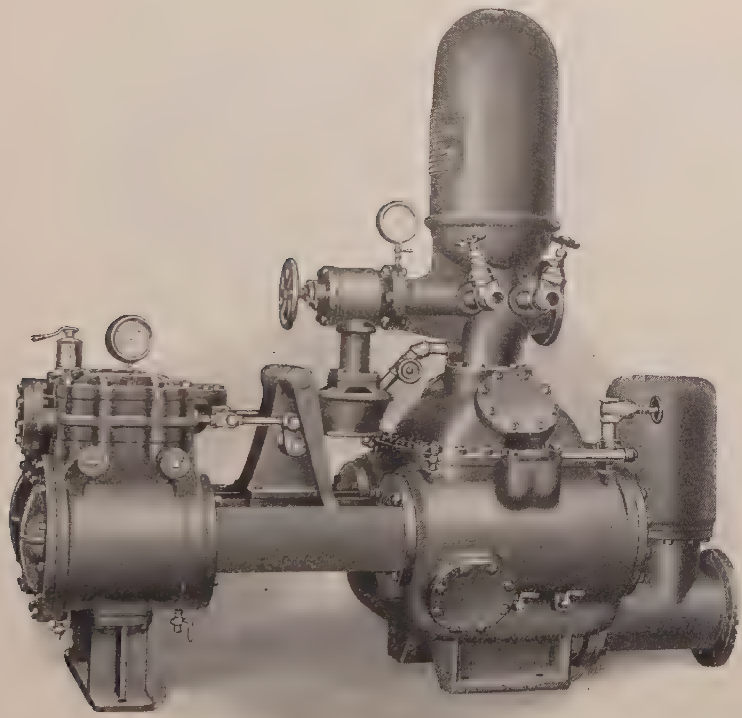


Fig. 95.

WORTHINGTON DUPLEX UNDERWRITER FIRE PUMPS

206. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Capacity Underwriter Rating				Pipe Sizes, Inches				Floor Space Inches
	Diameter of Steam Cylinders	Diameter of Water Plungers	Stroke	Gal. per min.	Fire Streams Number and Size	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
95	14	7¼	12	500	2-1⅛	140	70	3	4	8	6	110 x 41
95	16	9	12	750	3-1⅛	140	70	3½	4	10	8	111 x 48
95	18	10	12	1000	4-1⅛	140	70	4	5	12	8	111 x 51
95	20	12	16	1500	6-1⅛	160	60	5	6	14	10	136 x 53

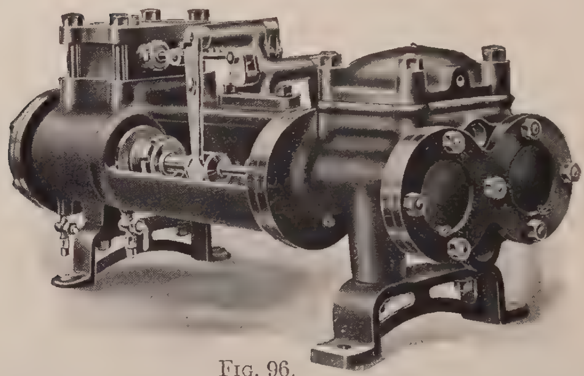


Fig. 96.

WORTHINGTON DUPLEX PACKED-PISTON PUMPS

Low Steam Pressure Pattern

With special ratio of cylinders for operation with steam pressures down to 6 lb. per sq. in.

224. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Diameter Liquid Pistons	Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boiler Pump will feed	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space Inches
96	4½	1¼	4	150	4.0	33	50	35	½	¾	2	1½	33 x 13
96	5¼	1¼	5	150	4.7	38	45	40	¾	1¼	1½	1	38 x 16
96	5¼	1¾	5	150	9.0	38	45	80	¾	1¼	1½	1	38 x 16
96	6	1¾	6	150	9.7	40	40	90	1	1¼	1½	1	42 x 17
96	6	2	6	150	13	40	40	120	1	1¼	1½	1	42 x 17
96	6	2¼	6	150	16	40	40	145	1	1¼	1½	1	42 x 17
96	6	2½	6	150	20	40	40	180	1	1¼	1½	1	42 x 17
96	7½	2¼	6	150	16	40	40	145	1½	2	4	3	45 x 22
96	7½	2½	6	150	20	40	40	180	1½	2	4	3	45 x 22
96	7½	2¾	6	150	24	40	40	225	1½	2	4	3	45 x 22
96	9	2¾	6	150	24	40	40	225	1½	2	4	3	47 x 23
96	9	3¼	6	150	33	40	40	315	1½	2	4	3	47 x 23
96	7½	2¼	10	150	23	58	35	220	1½	2	4	3	59 x 22
96	7½	2¾	10	150	34	58	35	325	1½	2	4	3	59 x 22
96	9	2¾	10	150	34	58	35	325	2	2½	4	3	61 x 23
96	9	3¼	10	150	49	58	35	460	2	2½	4	3	61 x 23

NOTE: These pumps may be combined with automatic receivers similar to the pumps listed in table, par. 228.

AUTOMATIC FEED PUMPS AND RECEIVERS

225. The Worthington Automatic Feed Pump and Receiver, shown in Fig. 97, and in section in Figs. 98 and 99, is designed to automatically drain heating systems, steam coils, drying cylinders, steam jackets, etc. It automatically returns to the boilers, with a very slight loss of heat, the water of condensation, obviating the necessity of employing steam traps or tanks. The principal difficulty in designing a device for automatically controlling the speed of a pump through the level of water in a tank is to secure a reliable **form of float.** It is impossible to make a hollow float which will stand varying temperatures under water pressure and remain tight. In place of the copper balls formerly used for this purpose, the Worthington apparatus employs the-open pattern bucket float

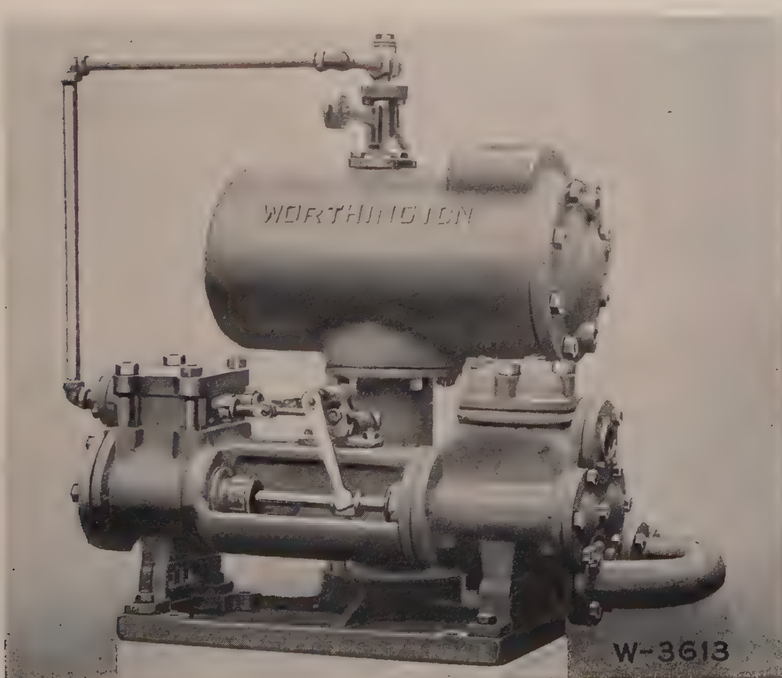


Fig. 97. Worthington Automatic Boiler-Feed Pump and Receiver.

with counterbalanced weight, as shown by sectional view in Fig. 99. Special attention is called to the fact that the bucket float and counter weight are placed in the interior of the receiver tank, doing away with all stuffing-boxes and outside levers and connections.

226. The **governor valve** is of the balanced type, directly connected with the float rod.

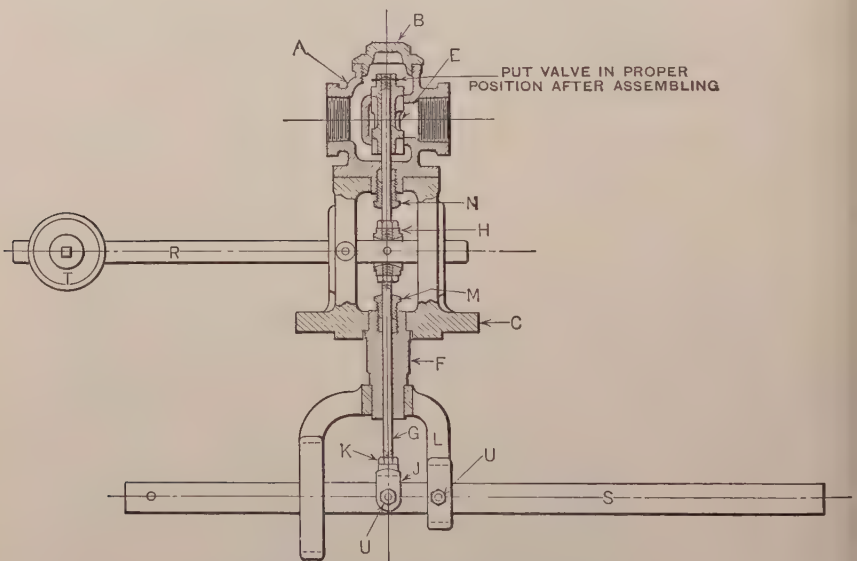


Fig. 98. Details of balanced valve used with Worthington Pumps and Automatic Receivers.

A—Valve Body.
B—Valve Cover.
C—Valve, intermediate.
E—Valve.
F—Lower Stuffing-box.
G—Spindle.
H—Crosshead.
J—Spindle Head.
K—Check Nut Spindle Head.
L—Lower Lever Yoke.
M—Stuffing-box Gland (lower).
N—Stuffing-box Gland (upper).
R—Lever (upper).
S—Weight and Bucket Lever.
T—Weight.
U—Lower Lever Pin.

227. When **using the automatic pump** and receiver for boiler-feed work, a cold-water make-up supply should be furnished to compensate for any loss which occurs through evaporation or other causes. If conditions are such that any grit or dirt is carried in the returns, the float should be cleaned out occasionally, as the extra weight of the foreign material collecting in it would interfere with its proper working.

228. TABLE OF SIZES AND DATA

Maximum Working Pressure: Steam End—150 lb. per sq. in.
Water End—200 lb. per sq. in.

Fig. No.	Size of Pump in Inches	Sq. Ft. of Radiating Surface Pump Will Drain	Capacity Gal. per Min. Boiler Feeding	Dia. Inches				Space Occupied, Inches		
				Steam	Exhaust	Water Inlet	Discharge	Length	Width	Height
97	4½ x 2¾ x 4	13,600	14	½	¾	2	1½	43	22	37
97	5¼ x 3½ x 5	24,000	24	¾	1¼	3	1½	47	26	42
97	6 x 4 x 6	33,900	34	1	1¼	3	2	55	27	43
97	7½ x 5 x 6	53,000	53	1¼	2	3	3	58	32	48
97	7½ x 4½ x 10	63,500	64	1¼	2	3	3	70	32	48

3 linear feet of one-inch pipe equals 1 sq. ft. of radiating surface.

229. The first two sizes are always fitted with bronze rods and bronze water pistons. On all other sizes these bronze parts are extra and only furnished when specially ordered.

230. Setting Up and Operating.—Before connecting steam pipe to the pump, be sure that the supply piping is thoroughly blown out so as to avoid the possibility of carrying any grit or dirt into the automatic valve. Be sure that the automatic-valve stem works freely and does not stick at any point.

231. Before starting pump, remove the head of the receiver and take out the blocking, which is placed at each end to hold the float and counterweight in place during transportation. Replace the head and allow receiver to fill with water to the top of the gage glass, at the same time holding up the automatic valve lever "R" so as to force down the Bucket "V" and cause it to fill. (See Figs. 98 and 99.) Do not forget to do this, as the perfect operation of the apparatus depends on this float being filled with water. Do not move the bucket counterweight; all adjustments should be made with outside counterweight "T." To change the water level in the receiver, move the automatic valve counterweight "T" in or

out as required. To raise the water level in the receiver, move the counterweight "T" in. To lower the water level in the receiver, move the counterweight "T" out.

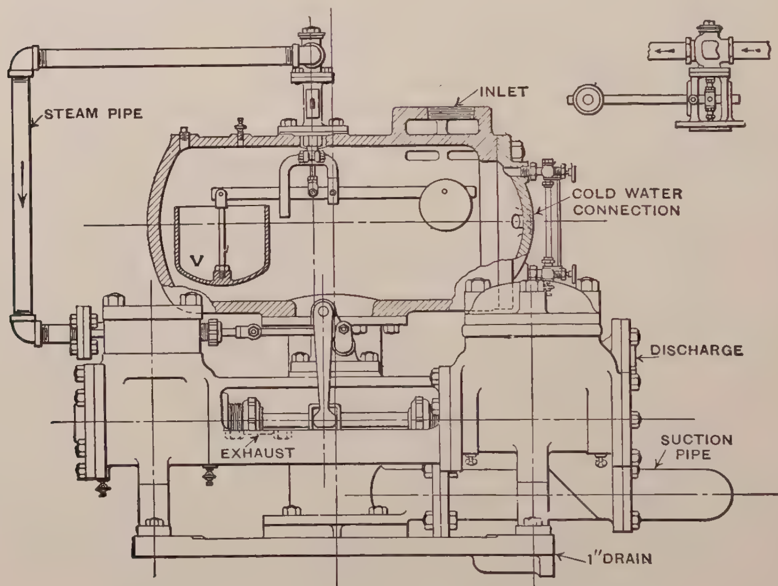


Fig. 99. Cross section of automatic receiver used with Worthington Pumps.

232. If for any reason the inside counterweight becomes loose, empty the bucket "V" and move the bucket counterweight until it just balances with the outside weight off.

233. There should always be a cold-water connection to the receiver for use in priming and in case of excessively high temperature. If the conditions are such that any amount of grit is brought in, in the returns, it may be necessary occasionally to clean out the float, as the extra weight will interfere with its proper working.

234. It is always advisable in piping up these machines to have a by-pass, so that in case of repairs the returns can be carried off without going through the receiver. The pump exhaust should be connected to the hot well, feedwater heater or atmosphere. A stop valve is also recommended, and should be placed outside of the

automatic steam valve, so that in case of accident to the automatic valve, steam may be shut off during repairs.

235. If it becomes necessary to remove the automatic steam valve, be sure first to remove the lever pins "U" and unscrew the lower spindle head "J" and check nut "K" from the spindle "G," then disconnect the yoke "L" by loosening clamp bolt. (See Fig. 98). Refer to the sectional view. In regard to the packing: Any first quality soft packing is all right for the piston rods in piston pumps. Every automatic feed pump and receiver is carefully tested before shipment, and should not be tampered with before setting up and starting.

236. In case of any difficulty arising that cannot be remedied on the spot consult the nearest Worthington office, stating the conditions under which the pump is operating, and information or suggestion will be cheerfully given.

WORTHINGTON VERTICAL AND SPECIAL PATTERN PRESSURE PUMPS

237. This book presents a number of lines of pumps which may be said to have been standardized. However, during the past eighty years many special patterns have been accumulated which make it possible to offer machines exactly proportioned for any service. Therefore, if in the preceding pages a pump is not found which in every respect precisely covers the requirements, write to Worthington giving details and character of the service, and drawings and specifications of suitable machinery will be furnished.

238. For instance, there are many situations where the floor space is limited or where for some other reason vertical machines are preferable. Worthington has many vertical patterns, and in Fig. 100 is illustrated a 20 x 6¼ x 18 simple pressure pump with Corliss steam valves, the water end being designed for 1000 lb. per sq. in. pressure, which is working in a rubber manufacturing plant; in Fig. 101 is shown a compound pump with slide steam valves, size 12 and 17 x 3 x 15, water end for 1500 lb. per sq. in. pressure; and in Fig. 102 is shown a large vertical triple expansion engine, size 16 and 25 and 42 x 20 x 24, water end for 200 lb. per sq.in. pressure. These examples will indicate something of the possibilities of pump-

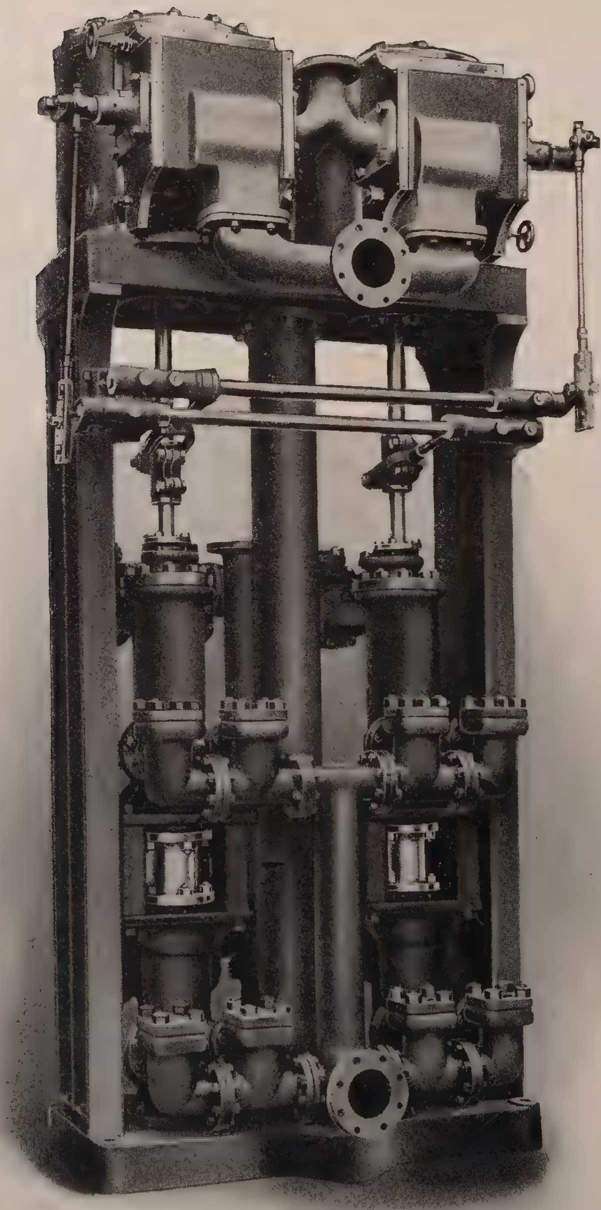


Fig. 100
Worthington Vertical Heavy-Pattern Pressure Pump

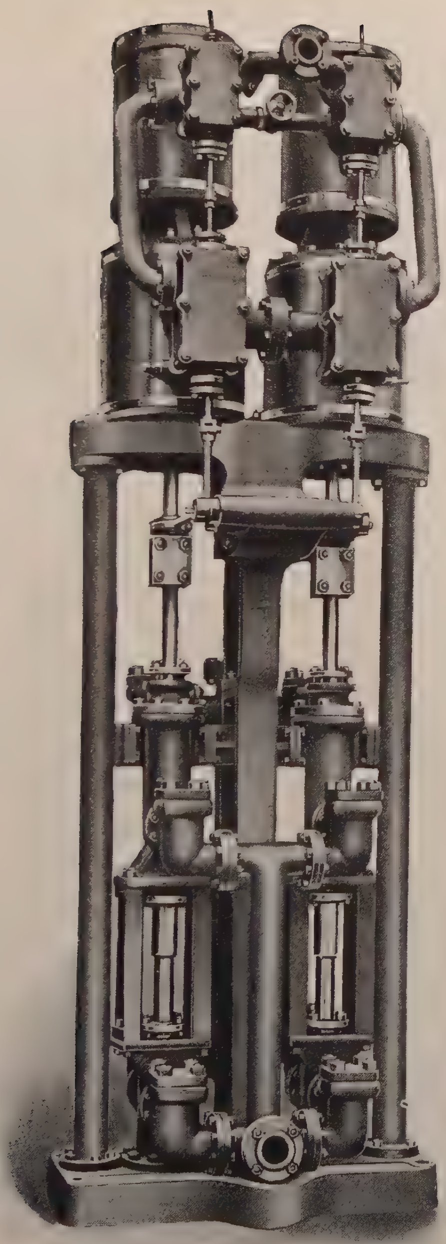


Fig. 101.

Worthington Vertical Compound Pressure Pump.

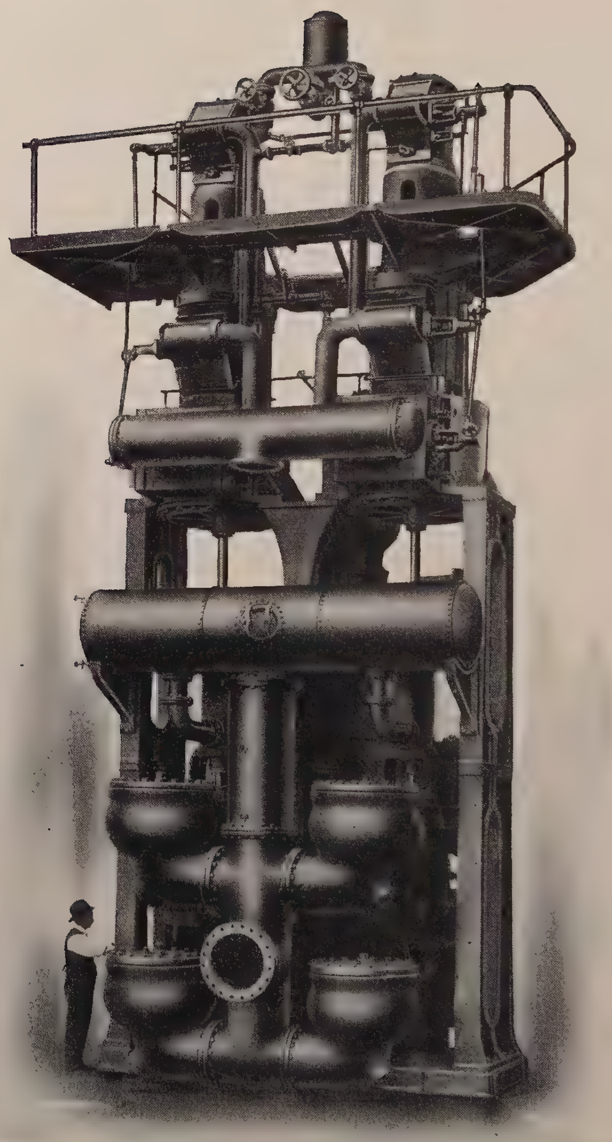


Fig. 102.

Worthington Vertical Triple-Expansion Pot-Valve-Pattern Pumping Engines.

ing with vertical machines where conditions render it necessary to do so.

239. The Worthington Engineering Department is always ready to make a study of any unusual problem. The Worthington plants are particularly well equipped to furnish pumping machinery for services which demand something out of the ordinary line; first, because no matter what the service, a suitable standard steam end pattern can always be supplied; and, secondly, because in the large majority of instances a water end can be made up by using patterns for standard valve pots, valve service, plungers, etc. making a special pattern only for the cylinder barrels.

240. Correspondence and consultation upon any difficult pumping problem which may be encountered are invited.

THE WORTHINGTON "SIMPLEX" PUMP
(Reg. U. S. Pat. Off.)

250. Features.—The notable feature of the Worthington Simplex Pump is its steam-valve gear, which is altogether the simplest, most durable and most reliable mechanism built for single direct-acting pump operation. The Worthington Simplex Valve Gear is a pronounced improvement over all earlier simplex valve gears and one of the strong features of this gear is its adaptability for operation with steam pressures up to 250 lb. per sq. in., and without cylinder lubrication if desired. This is of great importance in marine practice where high steam pressures are almost universally used.

251. Other **points of excellence** to be found in the simplex pump are: Minimum space requirement, minimum number of wearing parts, minimum number of stuffing boxes, pistons and joints to pack and keep tight, minimum amount of steam-cylinder condensation; pump makes full stroke at all times, will not short stroke under any condition of operation, which means minimum clearance spaces that must be filled with steam and consequently a material saving in steam consumption. A saving of 15 to 30 per cent in steam consumption is shown when comparing simplex and duplex pumps of equal capacity and pressure. When cut-off valves are applied to the simplex pump, a further saving of 8 to 15 per cent can be made. The simplex pump also is the best means for handling

hot water, gases, mixtures of liquids and gases or any service where the water cylinder may not entirely fill with liquid at each stroke.

252. Construction of Simplex Steam End.—The construction of the Worthington Simplex Steam End, Style "A" is shown in Fig. 103 and Fig. 104. The working parts are identical for both horizontal and vertical pumps.

253. The working parts of this Simplex **Valve Gear** consist of a main slide valve (1), which controls the admission of steam to the main steam cylinder (5), a steam-thrown plunger (3), which operates the slide valve (1), a small auxiliary slide valve (2), which controls the motion of the steam-thrown plunger (3). These three parts, together with the valve-rod connections (6 and 7) to the lever, which moves the auxiliary valve (2), make up the whole working mechanism of the valve gear. It is particularly adapted for operation with high-pressure steam, the small auxiliary valve (2) being the only working part direct connected to the valve rod, which valve, due to its small area, is subject to but little friction from the pressure of steam upon it, and consequently the strain and wear on the connections, which are encountered in some types, are eliminated.

254. The difficulties of **lubricating** at high temperatures being overcome by the design of the parts, the mechanism can be run without cylinder lubrication. It will be noted from Fig. 103 and Fig. 104 that the auxiliary cylinder carrying the steam-thrown plunger is at a right angle to the length of the main cylinder. This is so that the weights of plunger and main valve are supported when the pump is operated vertically, thus preventing any possibility of these parts dropping and short-stroking the pump, which can happen in types where auxiliary cylinder and main cylinder are in line with the plunger and valve operating vertically. The auxiliary steam valve in the Worthington Simplex Pump moves in line with the main piston rod, while the main valve operates at right angles to it.

255. Sizes up to and including pumps with 10-in. diameter **steam cylinder** are of the construction shown by Fig. 104. Sizes above 10-in. steam cylinders are shown by Fig. 103, the only difference

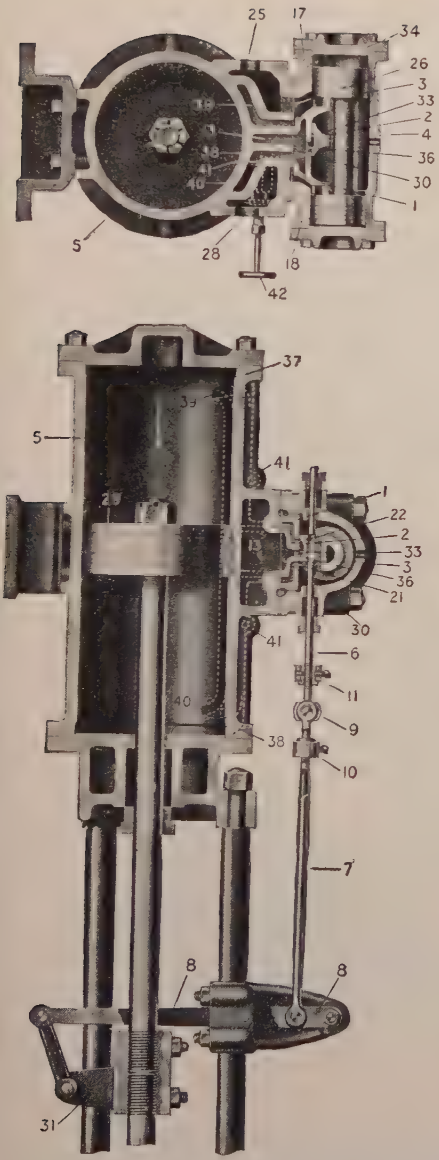


FIG. 103. Sectional view of Vertical Simplex Style "A" Steam End. Sizes with steam cylinders larger than 10 inches in diameter.

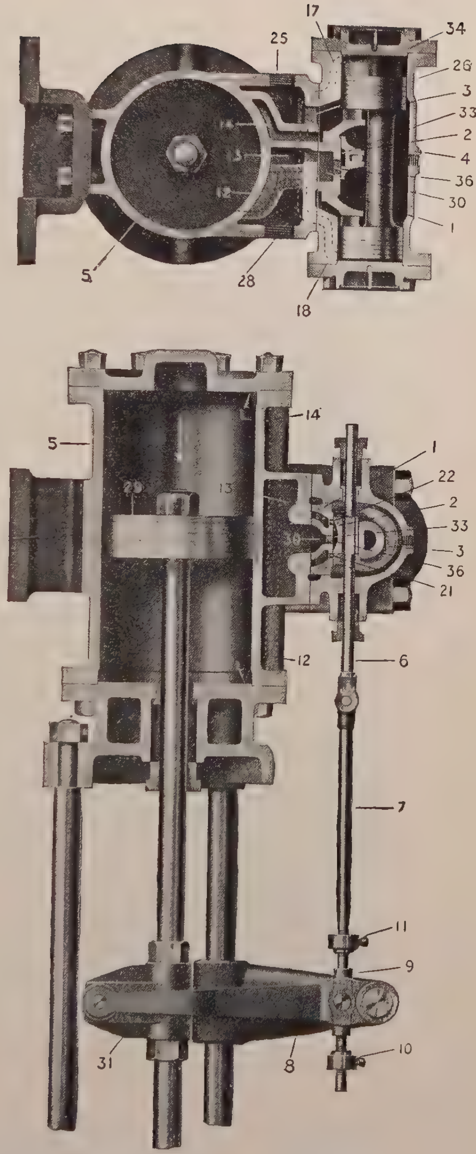


FIG. 104. Sectional view of Vertical Simplex Style "A" Steam End. Sizes having steam cylinders up to 10 in. in diameter, inclusive.

being in the steam port arrangement of the main cylinder. Steam cylinders larger than 10 in. in diameter have two ports side by side leading to each end of the cylinder, one of small area extending to the extreme end, called the "starting port" (37 upper and 38 lower), and the other of larger area extending to a point a little distant from the end, called the "main port" (39 upper and 40 lower). This main port also acts as the cushioning port at the end of the piston stroke in the same manner as in the case of the duplex

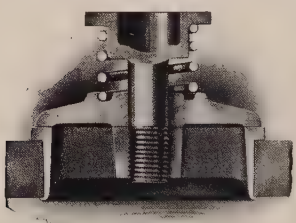


Fig. 105. Bronze Valve.

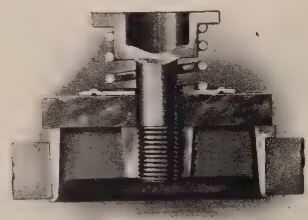


Fig. 106. Rubber Valve.

pump. Cylinders of 10 in. diameter and below have one port extending to each end (12 lower and 14 upper), these ports opening into the cylinder at a short distance from the ends (to provide

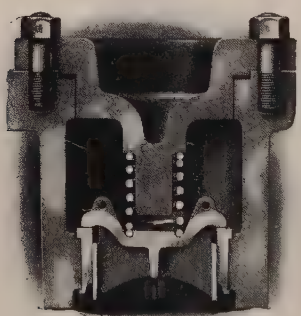


Fig. 107. Wing-guided Bronze Valve.

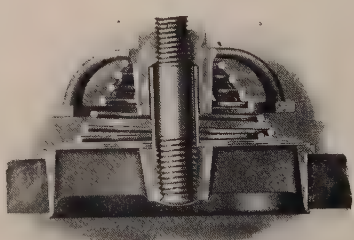


Fig. 108. "Kinghorn" Valve.

the cushioning effect at stroke end) with V passages reaching from the openings to cylinder ends, these V ports serving as starting ports in place of the independent ports of larger size pumps.

256. The **parts** making up the Simplex Style "A" **Valve gear** are assembled in the order shown by the sectional views, no adjustment for valves, plunger or any interior parts being necessary or possible.

The parts are so designed that they cannot be put together other than in the right way, which is a distinct advantage. It is to be noted that both main and auxiliary slide valves of the standard Simplex pump are without lap, and when on centers, the valves just cover the outer edge of the ports at each end of the cylinder; consequently steam does not cut off in the cylinder, but follows the piston the full length of the stroke. There are no dead points in Style "A" gear. The pump can always be started from whatever point it may have been stopped. The length of piston stroke is controlled by movable collars on the auxiliary valve rod, it simply being necessary to move these nearer together or farther apart, depending on the stroke length desired, and this can be done while the pump is in operation.

257. Worthington Simplex Pumps are furnished with **liquid ends** of either the horizontal or vertical types with inside-packed pistons for discharge pressure up to 300 lb. per sq. in. and in the horizontal type for pressures up to 3000 lb. per sq. in. The details of the liquid ends for the Simplex Pumps are very much the same as for the duplex except that they are fitted with only one double-acting piston, or two single-acting plungers. Discharge air chambers are always furnished with Simplex Pumps.

258. The **liquid valves** are one of the important features of a pump and must be of suitable design and of proper material to make the pump an efficient machine. Fig. 105 to Fig. 111 show types of valve service used in Worthington Simplex Pumps.

259. Simplex Standard Valves.—Standard valves for water pumps are shown by Fig. 105 and Fig. 106. The **bronze valve** is used for general service, and where hot water is to be handled. The **rubber valve** is of medium hard composition and is used as an alternative for service when the water is moderately warm or cold, or for light pressure.

260. Fig. 107 shows a flat-face bronze **wing valve** of the poppet type used for pot-pattern pressure pumps. These valves are guided by means of wings which work in the valve-seat opening; this being clear of ribs or other obstructions.

261. The **"Kinghorn" valve** consists of three very thin bronze disks placed loosely together, making a very strong and at the same time

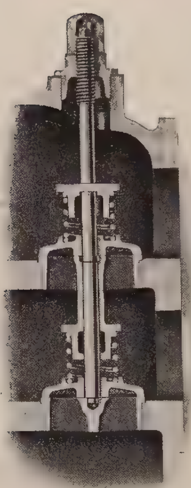


Fig. 109. "Bunge" Valve.
Disk type.

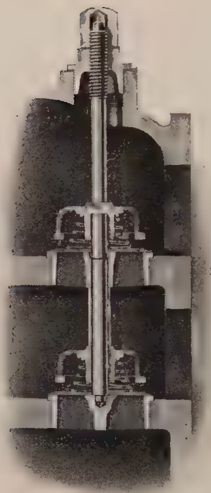


Fig. 110. "Bunge" Valve.
Kinghorn type.

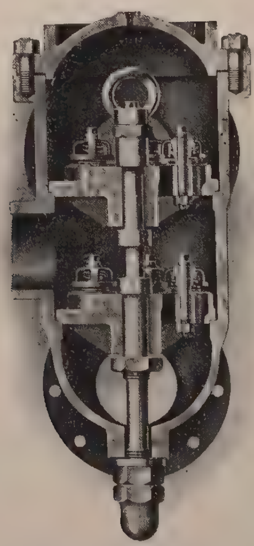


Fig. 111. Nested Valves.
"Kinghorn" type.

a very flexible and long-lived unit. This valve will seat absolutely tight and adapt itself to conditions as they are found. This type of valve is shown by Fig. 108.

262. It will be noted that all the foregoing valves have **seats** forced into the valve decks on a slight taper, with the projecting edge below the valve deck peened over to prevent the seat from working out.

263. Vertical pumps of the monochest type for ordinary service are fitted with valves of type shown in Fig. 105 and Fig. 106. For handling hot liquids, however, they usually have **"Bunge" valves** like Fig. 109. The suction and discharge seats are all in line and held in position by means of binding screws and caps adjusted from outside the cylinder. This arrangement prevents either valves, seats or springs from ever coming loose or ever getting adrift. The valves may be of bronze or rubber to suit service conditions or they may be of the "Kinghorn" type as shown by Fig. 110.

264. Vertical boiler-feed and pressure pumps of the valve-pot type are fitted with **nests of valves** of the "Kinghorn" type as shown by Fig. 111. The complete equipment for one pot, consisting of the suction and discharge valve plates with their valves, is carried on one central bolt which passes through the lower end of the pot and is securely held in place by means of a flanged and capped nut with proper gaskets. The entire valve service can be removed as a unit through the upper cover of the chamber, and every part is quickly and easily accessible for inspection or adjustment.

264a. The subject of liquid valves is covered more fully in par. 38 to par. 61.

265. Vertical Simplex Pumps.—The **purpose** of a vertical pump is to occupy small floor space, and when designed for use on shipboard or other location where weight is of importance, it must, of course, combine strength with the least possible amount of material. The work performed is in no way different from that of a horizontal pump, the construction being simply adapted to meet the limits of space. While both simplex and duplex pumps are built for vertical operation and are equally as serviceable as the same pumps made for horizontal use, simplex pumps are generally preferred for **marine work** on account of the smaller space required.

266. The Worthington Simplex, **Style "A"** Vertical Pump is a superior example of this type and in consequence is better known

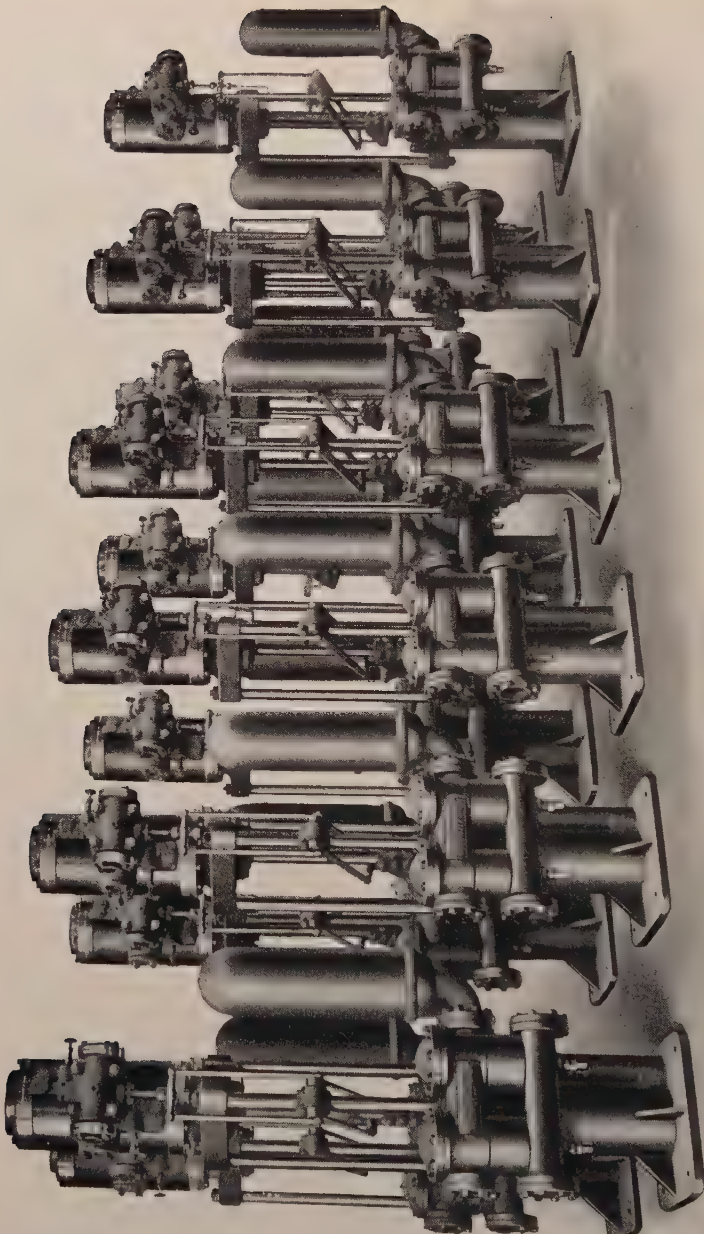


Fig. 112.

Group of Worthington Vertical Style A Piston Pumps Valve-pot Water End. Built for Marine Boiler-feed Service.

and more widely used for marine service than any other simplex pumps now built. A greater part of the **U. S. Navy ships** now afloat are equipped with these pumps and through the never-failing reliability of their performance they have become the standard against which naval engineers universally measure the operation of all marine pumps.

The accompanying illustrations are types of Worthington Vertical Simplex Pumps in which the compact, rigid, and light-weight construction is clearly shown. The steam ends are all of the well-known Worthington Style "A," in which the parts making up the construction are identical for both vertical and horizontal pumps.

267. The **liquid ends** are of two general types—one being the monochest (see Fig. 121) and the other the valve-pot type (see Fig. 123). In the former, the valves are located on two decks cast in the valve chamber and are accessible through large handholes on the chamber front. The valve service used in the pump is shown in detail in Figs. 105, 106, 109, and 110.

268. In the **valve-pot type,** each valve chamber is in the form of a cylindrical pot, and the valve service is arranged on plates, the complete equipment for each pot being removed as a unit through the upper cover of the chamber. A detailed description of this service will be found in par. 264. These pumps are types most used in marine practice, each having its own particular merits, the valve-pot type being favored for the higher discharge pressures on account of its greater strength per unit of weight, due to the cylindrical construction.

269. Both types of vertical pumps are built with either **bronze fitted** iron liquid ends, or liquid ends entirely of **bronze,** as may be required. The pumps listed in tables, par. 282 and, par. 284 cover sizes ordinarily built, but larger or intermediate sizes and other combinations of steam and liquid ends are built on order.

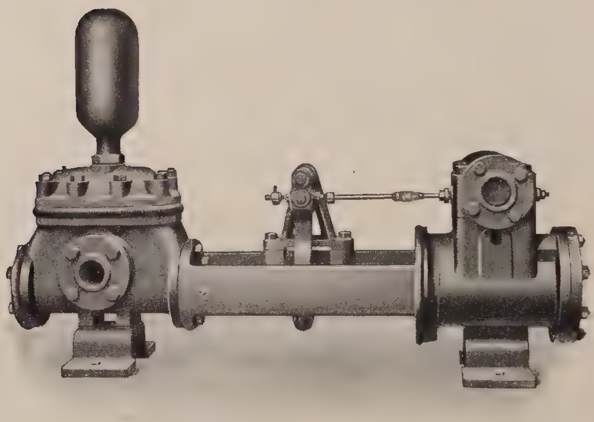


Fig. 113. Horizontal Simplex Boiler-feed or Pressure Pump, piston pattern. Style A.

WORTHINGTON SIMPLEX PACKED-PISTON PUMPS

Style A

Maximum working pressure: Steam end—250 lb.
Liquid end—250 lb.

275. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches: Diameter Steam Cylinders	Diameter Liquid Piston	Stroke	Max. Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boilers Pump will feed	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Space Floor Inches
113	3½	2¼	4	250	7	33	50	60	⅜	½	1	¾	29 x 8
113	4½	2¾	6	250	12	40	40	115	½	¾	1¼	1	39 x 11
113	5½	3¼	7	250	19	44	38	175	½	¾	1½	1¼	44 x 11
113	6½	4⅛	8	250	32	47	35	290	¾	1	2½	2	50 x 13
113	7½	4½	10	250	47	58	35	450	1	1¼	3	2½	59 x 15
113	8	5	12	250	60	60	30	565	1	1¼	3½	3	67 x 15
113	10	6	12	250	85	60	30	800	1¼	1½	4	3½	70 x 15
113	12	7	12	250	116	60	30	1100	1½	2	5	4	72 x 15
113	14	8	12	250	152	60	30	1450	2	2½	5	4	73 x 18
113	16	10	18	250	356	90	30	3400	2½	3	8	6	98 x 22

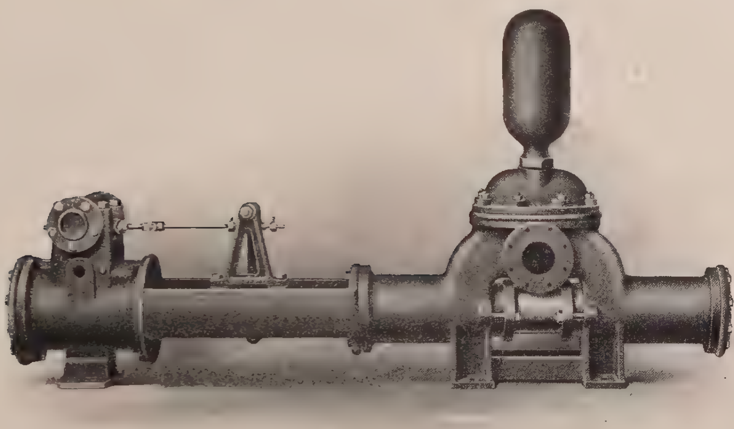


FIG. 114. Horizontal Simplex Center-packed Plunger Pump. Style A.

WORTHINGTON SIMPLEX CENTER-PACKED PLUNGER PUMPS

STYLE A

Maximum working pressure: Steam end—250 lb.
Liquid end—250 lb.

276. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation				Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinder	Diameter Plunger	Stroke		Gal. per Min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boilers Pump will feed	Steam	Exhaust	Suction	Discharge	
114	4½	2¾	6	250	12	40	40	115	½	¾	1¼	1	61 x 10
114	5½	3¼	7	250	19	44	38	175	½	¾	1½	1¼	66 x 10
114	6½	4⅛	8	250	32	47	35	290	¾	1	2½	2	70 x 13
114	7½	4½	10	250	47	58	35	450	1	1¼	3	2½	87 x 15
114	8	5	12	250	60	60	30	565	1	1¼	3½	3	96 x 15
114	10	6	12	250	85	60	30	800	1¼	1½	4	3½	101 x 15
114	12	7	12	250	116	60	30	1100	1½	2	5	4	106 x 18
114	14	8	12	250	152	60	30	1450	2	2½	5	4	113 x 18

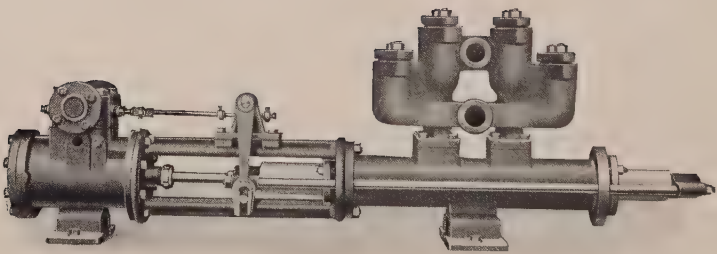


Fig. 115

Horizontal Simplex End-Packed Plunger Pump.
Pot-valve Pattern. Style A.

WORTHINGTON SIMPLEX END-PACKED PLUNGER PUMP

Pot-Valve Pattern—Style A

Maximum working pressure: Steam end—250 lb.
Liquid end—300 lb.

277. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure lb.sq.in. Good for	Capacity for Continuous Operation				Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinder	Diameter Plunger	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boilers Pump will feed	Steam	Exhaust	Suction	Discharge	
115	4½	2¾	6	300	12	40	40	115	½	¾	1½	1¼	64 x 11
115	5½	3¼	7	300	19	44	38	175	½	¾	2	1½	77 x 14
115	6½	4⅛	8	300	32	47	35	290	¾	1	2½	2	80 x 14
115													
115	7½	4½	10	300	47	58	35	450	1	1¼	2½	2	97 x 15
115	8	5	12	300	60	60	30	565	1	1¼	3	2½	107 x 17
115	10	6	12	300	85	60	30	800	1¼	1½	3½	3	113 x 20
115													
115	12	7	12	300	116	60	30	1100	1½	2	5	4	119 x 20
115	14	8	12	300	152	60	30	1450	2	2½	5	4	126 x 22
115	16	9	18	300	288	90	30	2750	2½	3	6	5	174 x 33

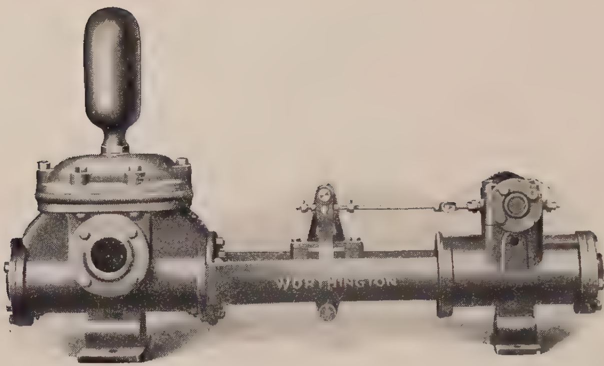


Fig. 116. Horizontal Simplex Tank or Light Service Pump. Style A. Cap and valve-plate type water end.

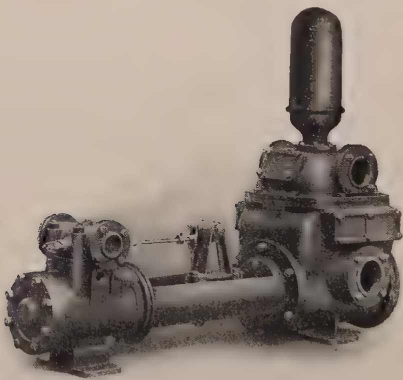


Fig. 117. Horizontal Simplex Tank or Light Service Pump. Style A. Close-clearance type water end.

Worthington Simplex Packed-Piston Pumps

Style A

WORTHINGTON SIMPLEX PACKED-PISTON PUMPS

Style A

Maximum working pressure: Steam end—250 lb.
Liquid end—see table

278. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation			Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinder	Diameter Liquid Piston	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
116	3½	3½	4	100	15	30	45	⅜	½	1½	1	28 x 10
116	4½	4⅛	6	100	24	36	36	½	¾	2½	2	46 x 14
116	5½	5½	7	100	52	43	39	½	¾	3	2½	46 x 14
116	6	6	12	100	77	54	27	¾	1	3½	3	66 x 15
116	7½	7	10	100	100	52	32	1	1¼	4	3	54 x 16
116	8	8	12	100	137	54	27	1	1¼	5	4	66 x 16
116	10	10	12	75	214	54	27	1¼	1½	6	6	74 x 22
116	10	12	12	75	308	54	27	1¼	1½	8	6	74 x 25
116	12	12	12	75	308	54	27	1½	2	8	6	76 x 25
116	10	10	18	75	320	81	27	1¼	1½	6	6	89 x 19
116	10	12	18	75	460	81	27	1¼	1½	8	6	90 x 25
116	12	12	18	75	460	81	27	1½	2	8	6	93 x 25
117	14	14	18	75	630	81	27	2	2½	10	8	99 x 29
117	14	16	18	75	822	81	27	2	2½	12	10	101 x 31
117	16	16	18	75	822	81	27	2½	3	12	10	101 x 31
117	18	18	24	75	1040	81	27	2½	3	12	12	120 x 40
117	18	20	24	75	1420	90	23	2½	3	14	12	120 x 40

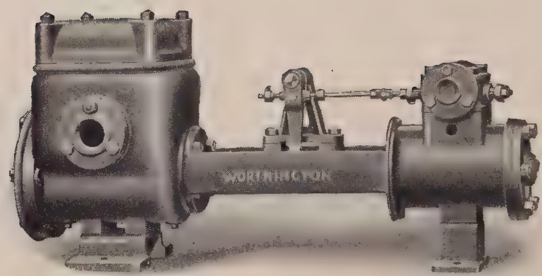


FIG. 118. Horizontal Simplex Double-acting Steam-heating Vacuum Pump. Cap and valve-plate type air end.

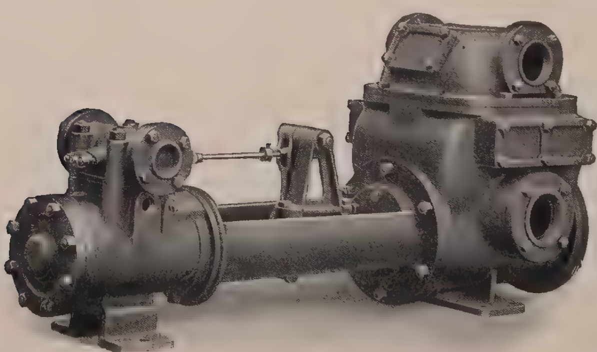


FIG. 119. Horizontal Simplex Double-Acting Steam-heating Vacuum Pump. Close-clearance type air end.

WORTHINGTON SIMPLEX STEAM HEATING VACUUM PUMPS

Style A

WORTHINGTON SIMPLEX STEAM-HEATING VACUUM PUMPS

Style A

279. These pumps are designed particularly for maintaining vacuum in connection **with steam-heating systems** for the purpose of insuring quick, positive and uniform circulation, a prompt removal of the hot water of condensation, prevention of water hammer and air binding, and increasing the economical operation of the heating plant.

280. The pumps are also equally well adapted for **general vacuum purposes** and are widely used for service in connection with surface condensers, sugar refineries, dye and chemical works and industrial evaporative processes generally.

281. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			*Maximum Operating Capacity Steam Heating Vacuum Service			Pipe Sizes, Inches				Floor Space Inches
	Diameter Steam Cylinder	Diameter Liquid Piston	Stroke	Sq. Ft. Radiating Surface Pump Will Handle	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
118	3½	5	4	7,000	55	83	⅜	½	2	1½	32 x 8
118	4½	6½	6	15,000	70	70	½	¾	3	2½	43 x 10
118	5½	8	7	26,000	80	69	½	¾	3½	3	47 x 16
119	6	6	12	18,300	100	50	¾	1	3	2½	65 x 13
119	6	8	12	32,600	100	50	¾	1	4	3½	65 x 16
119	8	10	12	50,900	100	50	1	1¼	5	4	68 x 20
119	8	12	12	73,400	100	50	1	1¼	5	4	69 x 22
119	10	12	12	73,400	100	50	1¼	1½	5	4	76 x 22
119	10	14	12	100,000	100	50	1¼	1½	6	5	78 x 30

*The above speeds and capacities are maximum.
Figured on a basis of 70 deg. F. room temperature and reasonably tight system.
For continuous operation we recommend a speed of 20 to 25 per cent. less than above ratings.

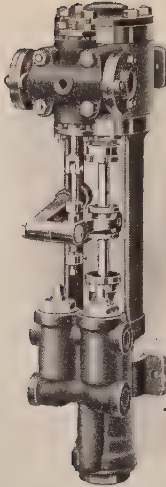


Fig. 120. Vertical Simplex Piston Pumps Style A, valve-pot type. Small sizes.

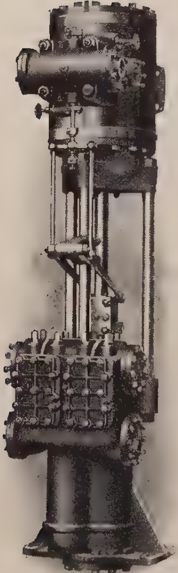


Fig. 121. Vertical Simplex Piston Pumps Style A. Mono-chest type.

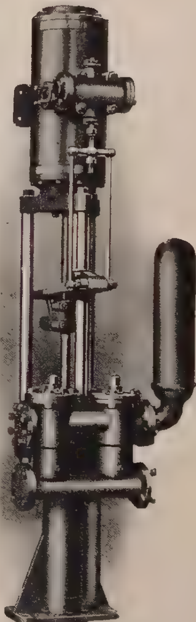


Fig. 122. Vertical Simplex Piston Pumps Style A. Valve-pot type. Intermediate sizes.

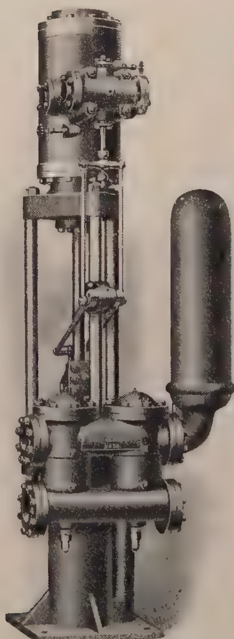


Fig. 123. Vertical Simplex Piston Pumps Style A. Valve-pot type.

Worthington Vertical Simplex Piston Pumps
Style A

WORTHINGTON VERTICAL SIMPLEX PISTON PUMPS

STYLE A

Maximum working pressure: Steam end—300 lb.
Liquid end—300 lb.

282. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation				Pipe Sizes, Inches				Space Occupied Without Air Chamber, Inches		
	Diameter Steam Cylinder	Diameter Liquid Piston	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Hp. Boiler Pump will feed	Steam	Exhaust	Suction	Discharge	Width	Depth	Height
VALVE-POT TYPE															
120	3½	1¾	4	300	4	33	50	40	⅜	½	¾	¾	9	12	32
120	3½	2¼	4	300	7	33	50	70	⅜	½	1	¾	9	12	32
120	4	2½	4	300	8	33	50	85	⅜	½	1	¾	9	12	32
120	4½	2¾	6	300	12	40	40	115	½	¾	1¼	1	10	15	41
120	6½	3½	8	300	23	47	35	220	¾	1	2	1¼	13	19	51
120	7½	4	10	300	37	58	35	350	1	1¼	2½	2	15	21	63
122	10	6	12	300	85	60	30	800	1¼	1½	3½	3	22	25	78
122	10	6	24	300	142	100	25	1325	1¼	2	3½	3	22	29	115
123	10	7	24	300	194	100	25	1820	1¼	2	4½	4	23	29	115
123	12	8	24	300	254	100	25	2400	1½	2	4½	4	28	34	128
123	14	9	24	300	320	100	25	3150	2	2½	5	4½	30	35	130
123	14	10	24	300	396	100	25	3750	2	2½	6	5	33	35	131
123	18½	13½	24	300	721	100	25	6800	2½	3	8	7	41	41	149
MONOCHEST TYPE															
121	8	5	12	300	60	60	30	565	1	1¼	3	2½	16	21	72
121	10	6	12	300	85	60	30	800	1¼	1½	3½	3	16	27	72
121	10	7	12	300	116	60	30	1100	1¼	1½	4	3½	17	25	78
121	12	8	12	300	152	60	30	1450	1½	2	4½	4	20	34	85
121	12	8	18	300	229	90	30	2175	1½	2	4½	4	23	30	105

WORTHINGTON VERTICAL SIMPLEX PISTON PUMPS

Style A, For Light Pressure or Tank Service

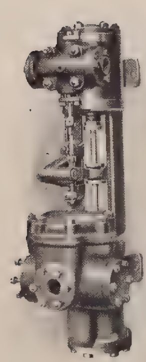


Fig. 124. Vertical Simplex Piston Pump. Cap and Valve-plate Type.

Maximum pressure:
Steam end—300 lb.
Liquid end—see table

283. The pumps of this series are built particularly for the limited installation space usually available on *shipboard* and are intended for handling the great variety of light-pressure work met in marine practice. In general design the pumps, and their many points of excellence as well, are the same as the line of pressure pumps, the main difference being that these combine larger pumping capacity with the same expenditure of steam, the water cylinders being larger in proportion to the steam cylinders, than in the case of the heavier pressure pumps.

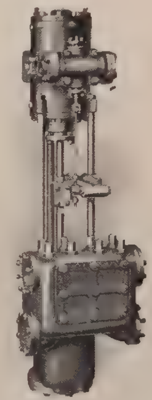


Fig. 125. Vertical Simplex Piston Pump, Monochest type.

284. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump in Inches			Max. Discharge Pressure Good for	Capacity for Continuous Operation			Pipe Sizes, Inches				Space Occupied Without Air Chamber, Inches		
	Diameter Steam Cylinder	Diameter Liquid Piston	Stroke		Gal. per min.	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	Width	Depth	Height
124	3½	4	4	75	19	30	45	⅜	½	2	1½	23	16	38
124	4½	5	6	75	56	36	36	½	¾	2½	2	23	17	41
124	4½	6	6	75	51	36	36	½	¾	3	2½	23	21	42
124	6½	7	8	75	84	43	32	¾	1	3½	3	26	23	61
125	6	7	12	50	105	54	27	¾	1	4	3½	24	26	72
125	7	8	12	50	137	54	27	1	1¼	4½	4	30	26	75
125	8	10	12	35	214	54	27	1	1¼	6	5	36	32	75
125	10	10	12	35	214	54	27	1¼	1½	6	5	36	32	77
125	10	12	12	35	308	54	27	1¼	1½	7	6	37	33	78
125	12	14	12	35	420	54	27	1½	2	8	7	41	38	81

WORTHINGTON ADJUSTABLE STEAM CUT-OFF VALVE
(Patented)

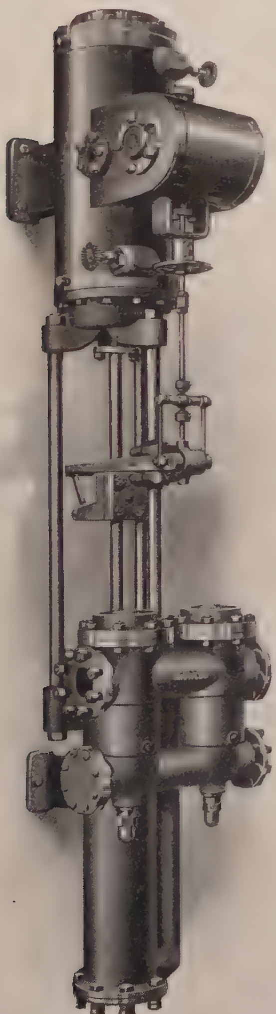


FIG. 126. Worthington Vertical Simplex Style A Piston Pumps. Long stroke, valve-pot water-cylinder type, with adjustable steam cut-off valve.

285. The pump illustrated in Fig. 126 is a refinement of the regular **Worthington** Simplex Vertical Style "A" Pump described in Par. 265, it having, in addition to the many advantages there enumerated, the unusual feature of an adjustable steam cut-off valve, the purpose of which is to increase economy in steam consumption by using the steam expansively. Another purpose that the valve accomplishes, due to the decreasing steam pressure after cut-off, is to gradually reduce the piston speed as it nears stroke ends. This allows the water-cylinder valves to seat quietly, thus preventing water shock and assuring a smooth reversal of piston stroke, which is brought about more effectively and with less expenditure of energy in this way than by trapping steam at the cylinder ends for cushioning, as is the case with ordinary direct-acting pumps.

286. The cut-off mechanism is very simple and adds but few parts to the regular Style "A" type of pump. It is adjustable by means of a graduated handwheel on the valve rod, and the amount of cut-off can readily be changed while the pump is in full operation. The valve will cut off at from three-quarters to full stroke, affording in this way an elasticity of

operation that makes it possible to meet economically the changes in load or steam pressure which are often encountered in actual service.

287. Fig. 127 shows **indicator cards** taken from both ends of the steam cylinder of one of these pumps, those in full lines being made while running without cut-off, and those in dotted lines, while the adjustable valves were set at the maximum, cutting off at about

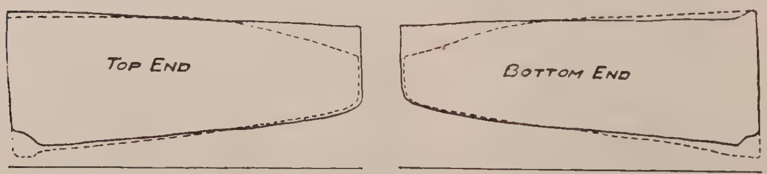


Fig. 127. Typical steam-end indicator cards showing the effect of cut-off.

three-quarters of the length of stroke. The reduced steam consumption and economical advantages of cut-off are plainly shown, the amount of saving in this particular case being practically 20 per cent.

288. Endurance tests of this type of pump, running at-125 ft. per min. piston speed and above, against 300-lb. per sq. in. water pressure, under boiler-feeding conditions, and without steam cylinder or valve lubrication, have been run repeatedly with most satisfactory results. Quotations on sizes adapted for any requirements of service will be promptly submitted.

SECTION VI

Installing and Operating Worthington Pumping Machinery

(Figures refer to paragraph numbers)

SECTION VI

Installing and Operating Worthington Pumping Machinery

1. **General Instructions.—Pumps should be located** in a light, dry, warm and clean room and should have the best possible attention. Do not overlook the importance of this suggestion. Exposure of the pump to an atmosphere filled with smoke, grit, moisture or dirt may cause rapid deterioration of the working parts. When it is necessary to locate pumps in pits, provision should be made to safeguard against floods. Motor-driven units should not be operated in damp or moist places, unless this condition has been especially provided for and the proper motor purchased. The pump should be located so as to be accessible for inspection during operation, and there should be ample head room to allow the use of an overhead crane, hoist or supporting structure of sufficient strength to lift the heaviest part of the unit.

2. The pump should be located as close to the source of supply as possible. If it can be done, place the pump below the level of the liquid in the suction reservoir so that the liquid will flow into the pump by gravity when the suction valve is opened, and the **pump** will thereby become **self-priming.**

3. A substantial **foundation** is absolutely necessary. It may be of any suitable material that is sufficiently rigid to support the pump permanently at all points, and to absorb any normal amount of vibration that may develop from any cause. Foundations may be of concrete, brick or stone. Concrete foundations built up from solid ground will prove the most satisfactory. In building the foundations, an ample allowance for grouting should be made.

4. **Foundation bolts** of the specified size should be accurately located according to drawings or templates, and each bolt should be surrounded by a pipe sleeve three or four diameters larger than the bolt (See Fig. 1). After the concrete is poured the pipe is held solidly in place while the bolt inside of it may be moved around to conform to the hole in the bedplate.

5. When a pumping unit is **mounted on steel work** or other structure, it should set directly over or as near as possible to the

supporting beams and walls. It must be supported in such a way that the base plate cannot be distorted and the alignment disturbed by any yielding or springing of the structure.

6. The **erection** of the pumping unit should be done by a thoroughly skilled and competent man. The base of the machinery must be placed upon the foundation and suitable **levelling** wedges of iron or steel placed at proper intervals to support the load solidly without springing. Adjust the wedges until the baseplate is level as indicated by an accurate level. Keep the level and all

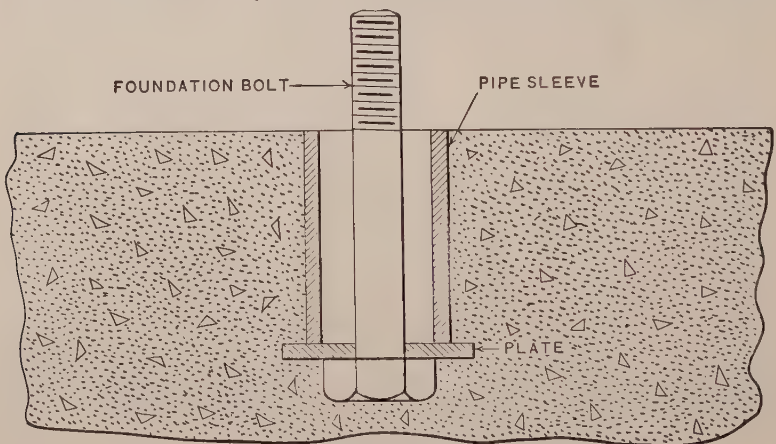


Fig. 1. Foundation Bolt and Pipe Sleeve.

surfaces very clean when levelling. After the grout is poured and fully set, the foundation bolts can be tightened up, using care not to distort the base or bedplate by the pull of the foundation bolts. Check alignment after pulling up on foundation bolts, as every baseplate is elastic, no matter how heavy it is, and will spring to a certain extent. The **alignment of all pumping units** must be accurately and permanently established if successful operation is to be secured.

7. On large size pumps, use **piping** with flanged joints and good grade gaskets. On small sizes the screw type of joint is satisfactory, providing the threads are well made up. If possible, keep all piping exposed and accessible. Care must be taken to properly support the suction and discharge piping so that no strain can be put on the pump from either its weight or expansion; therefore, before final connections are made to the pump, the **joints** must be made up so

that they will **meet exactly.** Pipe strains are very often the cause of misalignment, hot bearings, worn couplings and vibration. While piping is being laid and pump connected, extreme caution must be exercised to prevent foreign matter, sand, grit and rubbish getting into the pipe. Before finally connecting piping to the pump, wash it out thoroughly, and while the water or other liquid is passing through, rap the pipes vigorously to loosen the scale and oxide. Neglect of this precaution will result in damage to your pump.

8. Suction Piping.—The **suction connection** of pump may be identified from the discharge, as it is usually the larger of the two. The suction piping should be as direct and short as possible. It should be the full size as called for by the pump, and not have any branches. If the suction line is long the size should be increased. Care must be taken to guard against air pockets and not to install piping so it bows up and down, thereby permitting a possibility of entrapping air in the line. A horizontal suction line should have a gradual rise from well to pump. (See Fig. 4 and Fig. 6.) The suction pipe should be blanked off and a water test put on air leaks before starting up. The pipe should project into the well or source of supply deep enough to insure the pipe being well submerged when the water is at its lowest level. It should not extend too near to the bottom of the well where there is a possibility of the pipe becoming clogged up with foreign matter. Large **pipes** are usually **submerged** four times the diameter and small pipes two or three feet.

9. A good **arrangement of suction pipe** for a double-suction **centrifugal pump** is shown in B, Fig. 3. Figs. 2 and 3 illustrate how the suction piping should be installed to avoid air pockets in the suction line. When it is necessary to use an elbow in the suction line as shown by Fig. 5, the elbow should be located as shown in B. The flow is evenly divided and an equal quantity of liquid enters each side of the impeller, creating an hydraulic balance and preventing end-thrust. The piping arrangement shown in A should be avoided, as the flow is not evenly divided, resulting in more liquid entering one side of the impeller than the other, which destroys the hydraulic balance and causes end-thrust.

10. Never install a **positive-displacement pump,** either steam or power-driven, with a **suction line** arranged as shown in A, Fig. 6,

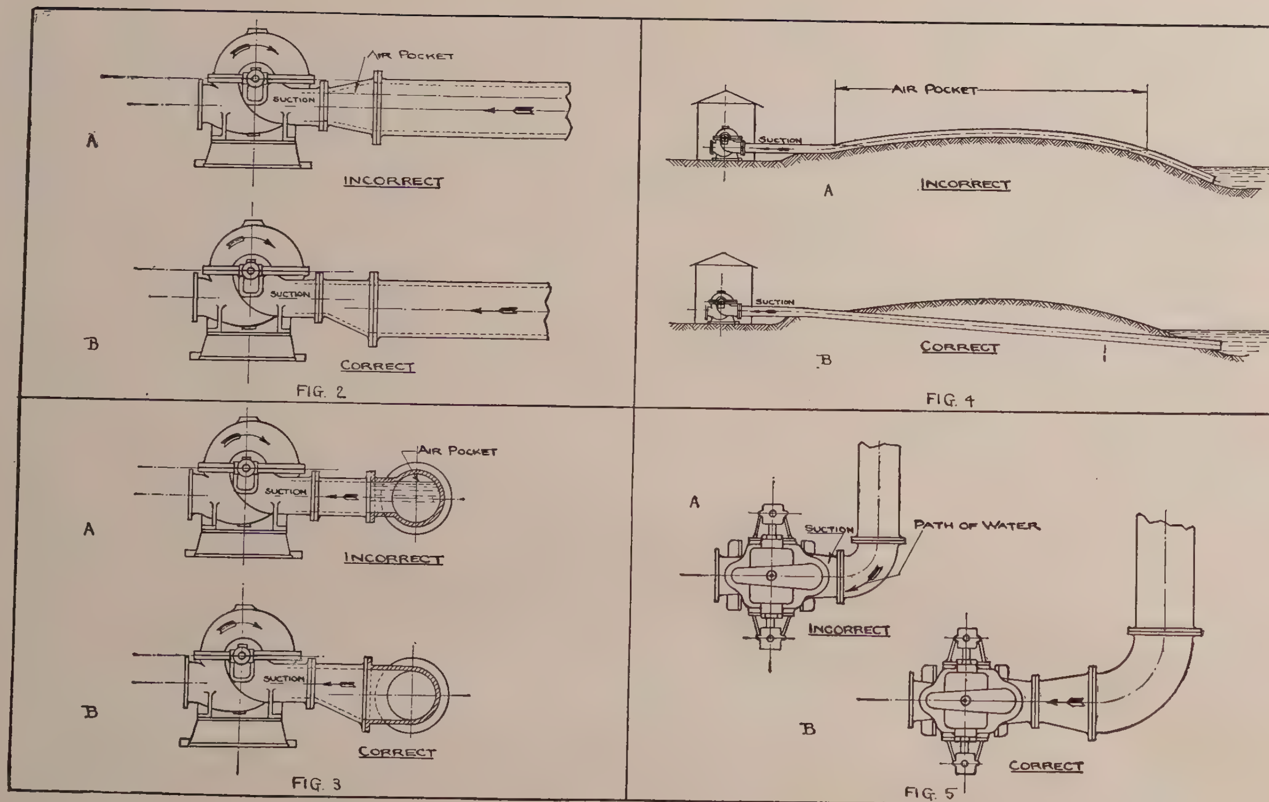


Fig. 2 to Fig. 5.

Methods of Connecting Suction Pipe to a Centrifugal Pump.

as trouble will be encountered due to the air pocket formed by the bow in the suction line. Always install the suction line as shown in B, Fig. 6, to avoid air pockets. A good installation with a vertical suction lift is illustrated by Fig. 9.

11. A **boiler-feed pump** taking its water from an open feedwater heater should be installed as shown by Fig. 7. The pump should be located so that the **head on the suction** is never less than the minimum given in the table.

12. Manufacturers of pumping machinery always specify the size suction pipe to be used with each size pump for the best results. For long suction pipes or high suction lifts, a vacuum chamber and a foot valve should be used. The **diameter** of the **suction pipe** should be such that the **velocity** of the water does not exceed 240 ft. per min. For estimating the cost of a suction line, the diameter may be calculated from the formula

$$D = \sqrt{0.1G}$$

where D is the diameter of suction pipe in inches and G the gallons per minute. For any special or complicated problem the purchaser should consult the manufacturer or a competent hydraulic engineer before deciding upon the size and arrangement of his suction.

13. For long or high suctions a **vacuum chamber** on the **suction side** of the pump is an advantage and is particularly recommended for simplex pumps, all fire pumps, and any pumps which are operated at high speed, especially for pumps of short stroke. For ordinary conditions centrifugal pumps do not require suction air chambers, but when the water contains a large percentage of air a vacuum chamber, to which is connected a vacuum pump or an air ejector for removing the excess air, is used.

14. The **location** of a **vacuum chamber** in the suction line of a direct-acting pump should receive careful attention. If the pump has an end suction the best location is on the dead end of a tee as shown by A, Fig. 8. If placed as shown by C, Fig. 8, much of the cushioning effect is lost. On large pumps a good location is on top of the run-around pipe. For pumps with the suction inlet on the side of the liquid cylinder, a good location for the vacuum chamber is shown by B, Fig. 8.

15. Too much attention cannot be given to the suction conditions of any pumping installation. Failure to observe **fundamental rules**

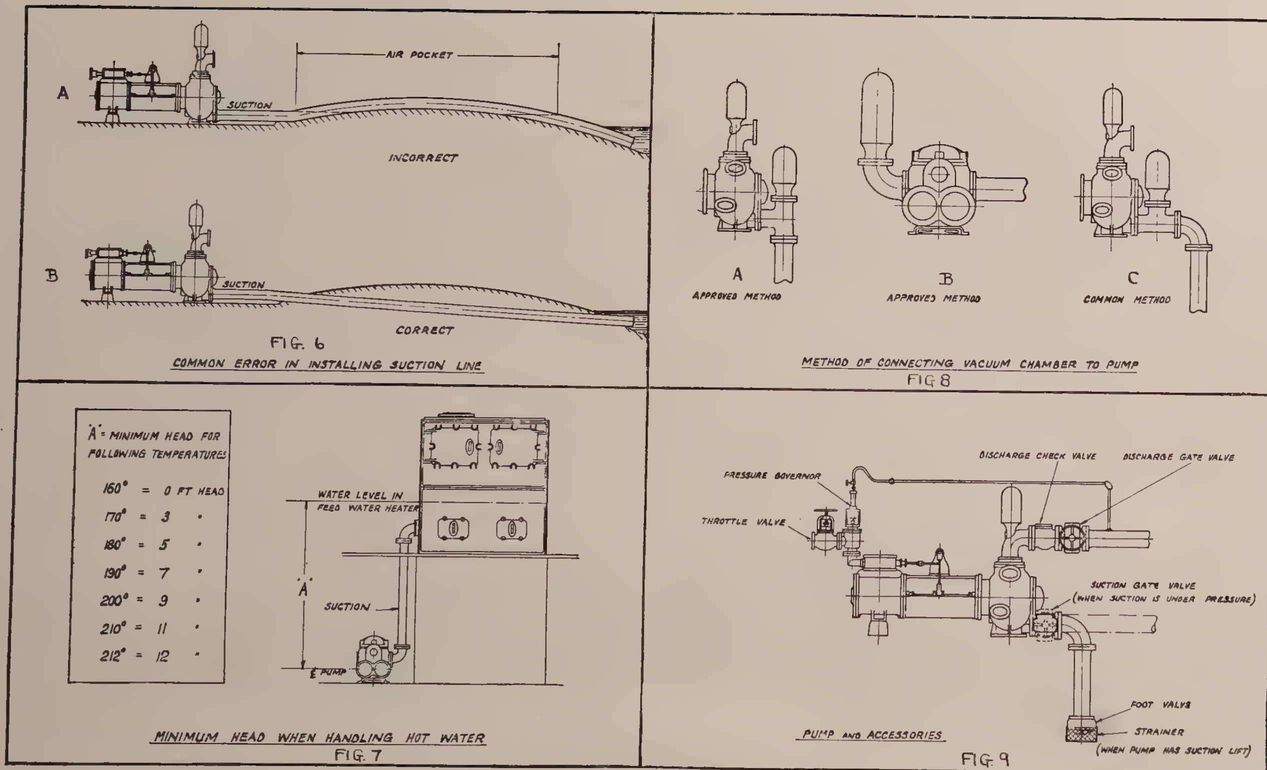


FIG. 6 TO FIG. 9

METHODS OF CONNECTING SUCTION PIPE TO DIRECT-ACTING PUMPS.

relating to the **suction** of a pump means the failure of the installation. It has been demonstrated by experiment that the pressure of the atmosphere at sea level (14.7 lb. per sq. in) will support a column of cold water at its maximum density, 33.83 ft. high. This experiment and demonstration was conducted by Torricelli in 1643 and was for the purpose of overcoming suction troubles with the pumps at the estate of the Duke of Tuscany. These pumps refused to "lift" the water more than 32 ft. Up to the time of Torricelli's experiment it was not accepted as a fact that the suction action of a pump is produced by atmospheric pressure acting on the surface of the water in the well, which forces the water into the liquid cylinder after the liquid piston had reduced the pressure in the cylinder below that of the atmosphere.

16. The **actual suction lift** that can be attained in practice is limited by the temperature of the liquid, the friction of the liquid through pipes and fittings and leakage of air through joints of the suction pipe. There is also a reduction due to velocity and entrance heads. As the altitude increases the atmospheric pressure decreases, resulting in a decrease of the possible suction lift.

17. As the **temperature** of the liquid increases, the possible **suction lift** decreases. For any temperature there is a definite pressure at which a liquid will vaporize. Water vaporizes at 212 deg. F. when under atmospheric pressure. If the pressure on the water is reduced, the water will vaporize at a lower temperature. If hot water is to be pumped and such a lift is attempted that the pressure on the suction side of the pump is below that at which the water will vaporize, vapors will be formed and the suction will fail. Hence, when **hot water** is to be **pumped** there must be a positive head on the suction side of the pump to prevent this vaporization.

18. The permissible suction lifts and the necessary positive suction heads for different altitudes and different temperatures of water are given in table, par. 20. Where the values are preceded by the minus (—) sign suction lift is indicated; where the values are preceded by the plus (+) sign a positive head is indicated. The values given in the table are for water only. Liquids such as gasoline, alcohol, crude oils, tar, molasses, etc., must be considered separately and each case decided upon according to the character of the liquid to be pumped.

20. An analysis of the **effects of suction lift** and suction head on the performance of centrifugal pumps will show clearly the importance of this subject. These remarks will apply in general to positive-displacement pumps. It has already been shown that suction lift automatically subtracts from delivery head with a given impeller. There is also a limit at which this can be done without affecting the capacity of the pump.

TABLE 20—SUCTION LIFTS

Minimum Allowable Head in Ft. on Suction	Temperature of Water in Degrees F.															
	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	210
At sea level	−22	−20	−17	−13	−13	−11	−8	−6	−4	−2	0	+ 3	+ 5	+ 7	+10	+12
At 2000 alt.	−19	−17	−15	−13	−11	− 8	−6	−4	−2	+1	+ 3	+ 5	+ 7	+10	+12	+15
At 4000 alt.	−17	−15	−13	−10	− 8	− 6	−4	−1	+1	+3	+ 5	+ 7	+10	+12	+14	....
At 6000 alt.	−15	−13	−11	− 8	− 6	− 4	−2	+1	+3	+5	+ 7	+10	+12	+14	+16	....
At 8000 alt.	−13	−11	− 9	− 6	− 4	− 2	0	+3	+5	+7	+ 9	+12	+14	+16	....	....
At 10,000 alt.	−11	− 9	− 7	− 4	− 2	0	+2	+4	+7	+9	+11	+14	+16	+18	....	....

21. It is of course fully understood that atmospheric pressure forces the liquid into the suction pipe where a lower pressure exists. This atmospheric pressure must not only overcome all pipe and entrance losses but the losses in the suction nozzle of the pump and the eye of an impeller as well as the reduction in pressure due to the increase of velocity in the impeller. The moment the liquid is picked up by the vanes of an impeller it has a higher velocity than before and this velocity also detracts from the atmospheric pressure. Since all these losses and velocity changes are dependent on the quantity flowing, it is clear that **suction lift affects the maximum capacity** that can be obtained from a **centrifugal pump.** Up to a total lift of 15 ft. these effects are not noticeable on the single-stage volute pumps for the usual heads. However, with high heads per stage requiring larger normal capacities the pump may not obtain the 100 per cent normal capacity as a maximum, which failure will cause the characteristic curve to break down suddenly into a straight vertical line. This means reduced output for the pump or it may require the use of a larger pump at reduced efficiency.

22. The Francis-type impeller with lower pick-up or absolute velocity at inlet of vane will help to reduce the effects of high lift to some extent; but without having all the data to check back with the impeller, it is not possible to make a definite statement as to

what **maximum lift** may be attained by a given pump. In some cases suction lifts of 20 to 24 ft. have been successfully and economically employed. All such cases should be referred to the manufacturers for analysis.

23. Whenever a pump operates at the **critical capacity point,** cavitation or erosion of vanes and side walls is set up, due to shock and the separation of air and gases from the liquid. It is recognized that this critical point corresponds roughly with the boiling point of the liquid at the existing vacuum or pressure existing at the vane tips.

24. For higher lifts the slower speed type of pumps with their longer impeller vanes are desirable, and if operated at or near the critical point of capacity will have the least effects on wear and erosion of the vane tips. Summing up the suction-lift problem, we find that much depends upon the total head, the type and speed of the impeller.

25. Whenever a **pump is located below the source** of supply or when it takes liquids from a system of piping under pressure it is necessary to place a gate valve in the suction line close to the pump. It is then possible to open the pump for inspection, cleaning or repairs without first draining the pressure suction system, and also to control the capacity to a certain extent on the suction side of the pump.

26. If a pump is called on to **pump light volatile liquids** such as gasoline, benzine, naphtha, etc. which vaporize rapidly in a vacuum, it is, of course, necessary to provide sufficient head on the suction to prevent such vaporization in the suction line or pump passages.

27. Hot water must always flow to the pump under a positive suction head. As this condition is closely related to the installation of boiler-feed pumps, the two subjects will be treated together.

28. In the preceding paragraphs the bad effects of operating a pump at the critical capacity point were brought out. We find this same **critical condition** applies to **boiler-feed pumps** handling hot water as the water temperature is at or near the boiling point at atmospheric pressure. Under these conditions it is necessary to arrange the pump with a positive head on the suction so that the absolute pressure existing at the pick-up part of the vane entrance is a few feet greater than that due to the temperature of the water.

29. Here again **stage pressure** as well as speed **determines** the amount of **suction head** required at a given temperature. This also leads to the adaptation of the double-suction impeller type of

pump as allowing of the least suction head for given temperatures and stage pressures. Roughly speaking a steam turbine-driven multi-stage boiler-feed pump of 160 to 180 ft. per stage and 212 deg. F. water requires a positive head of 8 ft. on the suction for normal capacity. If an over size, or lower-speed pump is used this head can be reduced. With very small pumps the conditions are not so accentuated.

30. In some power stations a low-head closed **heater is used** with a **back pressure** on the water to give the same effect as a higher head of water. Actually the effect is not the same, as the water has a greater tendency to vaporize throughout its mass than if an equal pressure was obtained by means of a greater depth of water column. An arrangement of this kind also has the disadvantage of a more rapid change in the temperature of the water in the event of a change in either the back pressure or the water level in the heater.

31. If **vaporization** is allowed to occur, a solid body of water is not available for the first-stage impeller and the pump quickly becomes vapor bound, and is thrown out of hydraulic balance. The unbalanced pressures result in burned out thrust bearings or thrust disks as the case may be, and the unit is entirely out of commission.

32. Another cause leading to vaporization while not due to suction conditions can be considered at this time, and that is the closing of all the check or stop valves in the discharge line. During this period the **pump** is simply **churning** the **water,** which raises the temperature so that a vaporization condition soon exists.

33. The foregoing analysis emphasizes the necessity of using the greatest care when installing the suction piping for a pump. particularly if it is for boiler-feed service. The utmost care may be used in selecting and building a pump, but unless equal care is used when installing, it will prove a failure in operation.

34. A **foot valve** on the suction pipe increases the friction, but when one is installed for convenience in priming, or to keep the pump primed when idle, it is recommended that a size be selected with a **port area** equal to at least twice the area of the suction pipe.

35. A **strainer** should be installed on the suction pipe to protect the pump from getting any foreign material into the impeller or valves. It is very important to take every possible precaution to protect the pump from becoming clogged up with foreign matter, as serious damage often ensues. A **strainer** with a net **area** of

three to four times the area of the suction pipe should be used. The net area is understood to mean the clear and free opening through the strainer. If the strainer is likely to become frequently clogged up, an accessible place should be selected for the suction pipe to facilitate cleaning the strainer. For large pumps removable screens should be placed at the entrance to the suction well.

36. When the liquid flows to the pump under a head, a **gate valve** should be placed **in the suction line** for use when occasion requires.

37. When a **stop valve** is located **in the suction line,** a small spring relief valve must be installed between the pump and the stop valve to prevent pressure building up on the suction system. This relief valve should be set at 25 lb. per sq. in. pressure.

40. Discharge Piping.—Standard practice among pump manufacturers is to furnish their pumps with all discharge passages connected to one common opening to which the purchaser connects the discharge pipe. The discharge line should be of ample size and with as few bends and fittings as possible in order to reduce the friction head to the lowest consistent point, since excessive friction head is merely wasted energy. The **velocity** in the discharge pipe should not exceed 300 ft. per min. for the best results. For the purpose of estimating the cost of a discharge line the **diameter** of the pipe may be approximated from the formula:

$$D = \sqrt{0.08G}$$

where D is the diameter of pipe in inches and G the gallons per min.

41. In order to provide an efficient shock absorber in the discharge line, an **air chamber** is provided, as shown by Fig. 9. Discharge air chambers are a necessity for single-acting pumps of either the simplex or duplex type and for crank and flywheel pumps of any type. A discharge air chamber is not necessary for duplex double-acting low service pumps where the discharge pressure does not exceed 75 lb. per sq. in or for small duplex double-acting pumps for general service (10 by 6 by 10 and under).

42. The **volume of the air chamber** should be six to eight times the displacement for single direct-acting or for crank-and-flywheel pumps. For duplex pumps the volume should be three to four times the displacement. Air chambers should always be provided with gage glasses and suitable air cock on top as a means of releasing excess air and keeping the water level at the proper height.

43. The discharge pipe should be installed with a **check valve** and **a gate valve** near the pump outlet. The check valve protects the pump from excessive pressure and prevents a centrifugal pump from running backward as a water wheel in case the prime mover trips out and stops without the operator's knowledge. A gate valve is necessary for starting up a centrifugal pump and for repairing all types of pumps, or during idle periods. The gate valve should be installed on the outboard side of the check valve.

44. A **relief valve** should be placed next to the pump, in the discharge pipe of every power-driven positive displacement pump. This valve is for the purpose of protecting the pump against breakage caused by the closing of the main discharge valve, thus increasing the pressure above the maximum for which the pump is designed. The relief valve should be frequently tested to prevent sticking. The spring should be set so as to open at a pressure slightly in excess of the maximum operating head.

45. A **priming pipe** connected to a supply above the pump is a convenience for quick starting, and a necessity for a centrifugal pump, a fire pump and most large positive displacement pumps. Centrifugal pumps may be primed by means of an air or steam ejector located on the pump casing.

46. To **prevent freezing,** drain your pump by opening all plugs and cocks provided for the purpose. Non-freezing mixtures if used must be absolutely non-corrosive and free from grit.

47. The frequent **inspection** of packings and the regrinding of pump valves and other valves is recommended, especially on a new system, also that your pump be looked over weekly.

48. **Pumps and Accumulators.**—When a **pump** is to be **used** in connection with **an hydraulic accumulator,** the water pumped should be returned to a suction tank and used over and over. This effects economy in water. It also permits of treating the water so that it will not corrode the interior of the pump, the pipe line, or the operating valves. This suction-supply tank should be baffled to prevent air being carried into the pump suction. The tank should be of such area that the water will not become heated sufficiently to injure valves or packings; and a close fitting (but not air-tight) cover should be on tank to keep the water free from dust and dirt.

49. **All water returned** to the **suction-supply** tank should be

filtered or strained through a number of layers of filter cloth to eliminate all grit, oxide from the interior piping, sand, pipe cuttings, etc. which will quickly cut the plungers, pistons, and valves, doing more damage to the machinery than years of proper use. Treat the water supply with some softening and lubricating agent, being careful to select such agent as will not cause the water to affect the operating valve or pump packings.

50. It is desirable in connection with accumulation that the fluid flow to the pump under a slight head; but, where necessary, the machine can be operated successfully with a suction lift not exceeding 8 ft. total.

51. No butterfly valve (or other valve) should be installed in suction line to pump; but in the discharge line between a power-driven pump and the accumulator a Worthington **automatic by-pass** and check valve should be installed. Full information will be furnished upon request.

52. It is recommended that the pump be operated for some hours under no pressure with all valves open, and with the filter at work, if a filter is included, to dislodge and eliminate as far as possible all grit, scale, and oxide.

53. Noiseless Operation.—In the case of pumps furnished for **house service** or for any service where noise is objectionable, the pump is not to be held in any way responsible for noise caused by air in the suction or discharge pipes, or for noise due to improperly designed or installed piping. Vibrations and pulsations incidental to the operation of pumping units are easily transmitted from the pump foundations or pipes through piers and columns of buildings, especially those of steel-frame construction. To avoid transmission of such vibrations and pulsations, care should be taken that pump foundations, and piping leading to and from pumping units, are not in masonry or metallic contact with footings of piers or building foundations, with concrete or tile floors or with structural steel work. The use of **flexible suction and discharge connections** to the pump is strongly recommended.

INSTALLING CENTRIFUGAL PUMPS

60. Alignment of Centrifugal Pumps.—Usually it will not be necessary to take 12-inch and smaller centrifugal pumps off the

bedplate while levelling up. Larger sizes should be removed from the bedplate until it is levelled up, unless the erection is supervised by an experienced centrifugal pump erector. After the bedplate has been levelled and the foundation bolts pulled up, thoroughly clean the pads of the base and of the pump and prime mover. Place the pump and prime mover on the bedplate and insert the holding down bolts, but do not tighten them. Remove the bearing covers and top shell of the pump bearings and place an accurate level on the shaft. Tighten the holding down bolts on the pump maintaining it level. If necessary, use shims to do this. With the pump set and holding down bolts tightened, align prime mover to pump by method as shown by Fig. 10 and 11.

61. In **truing up a coupling,** clean off all paint and burrs on the coupling. Use a straight edge and a set of feelers or thickness gage. When the couplings are perfectly true on both the faces and outside diameter, and are of the same diameter, the exact alignment will show that the distance between any points on the faces are the same at all points and a straight edge will lie squarely across the rims of both halves at any point. If the faces are out of parallel, the thickness gage, or feelers, will show a variation at different points. If one coupling is higher than the other the amount may be determined by the straight edge and feelers.

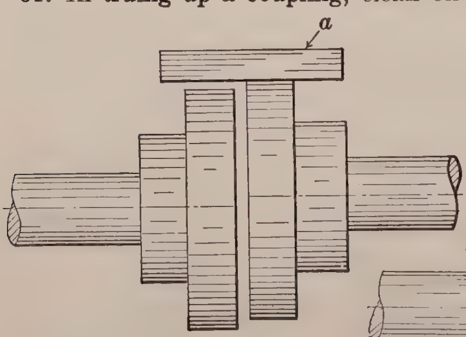


Fig. 10.

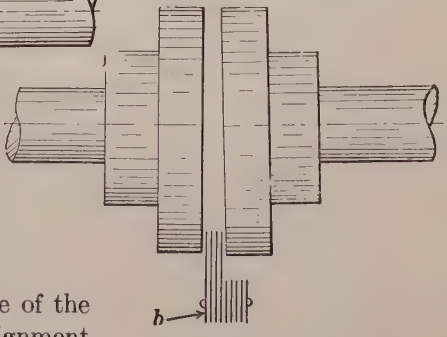


Fig. 11.

Method of truing up couplings.

62. If the couplings are not true or of the same diameters, proper allowance must be made. **To check the trueness** of the coupling, chalk four marks 90 degrees apart on the circumference. Hold the

half coupling on the pump stationary and revolve the other half on the turbine or motor one quarter of a turn, or 90 degrees at a time, and check the alignment each time. Particular care must be taken to keep all end play on the revolving coupling in one direction. The driver (turbine or motor) half coupling should now be held stationary, the pump coupling revolved and the four points on pump coupling compared with one point on stationary coupling. If any variation is found in the coupling, proper allowance must be made in aligning the machine.

63. When pumps are driven by turbines an allowance must be made for the fact that the turbine will expand as it is heated and will rise. To **allow for that expansion** when lining up cold, the turbine should be set 0.006 in. to 0.012 in. low, depending on its size. In any event, when the **turbine is lined up** cold the alignment should be checked when hot. Further checking should be made after the unit has been in service several hours.

64. No heat allowance is made for **electric motors.** However, the **motor** should be operated alone if possible before aligning the pump so as to determine the magnetic center of the rotor. If this is not possible the rotor of motor should be pulled over and pushed back to determine the collar clearances, and then the rotor placed in midposition for aligning.

65. The **clearance between** the faces of **couplings** should be set so they cannot strike, rub or put a pull on either machine. A clearance of one-quarter inch is usually allowed on small machines.

66. Do not put coupling bolts in until piping is complete and the turbine or motor tried out for correct direction of rotation. All pipe flanges must come true. Check the alignment after the machine has been completely piped up. When checking the alignment make sure the coupling bolts have been removed.

67. All prime movers shipped to our factory are mounted on bedplate and lined up. The factory does not **dowel** either the **pump** or **prime movers.** This may be done after the machines are completely erected and ready for service.

68. Piping.—If a thrust bearing is supplied, make sure that the **jacket cooling-water pipe** is connected so that the supply enters the bottom and discharges from the top. For the purpose of observing whether the jacket water is flowing and for the regulation of the

amount, it is good practice to pipe the discharge jacket water so it will give a free flow into a funnel connected to a drain.

69. All **drain connections** should be piped to sump pit or suction well so that the drain water will be properly carried away.

70. **Piping to each gland** should have a valve to control the amount of water necessary to feed and seal each gland and to permit enough seepage to lubricate the gland.

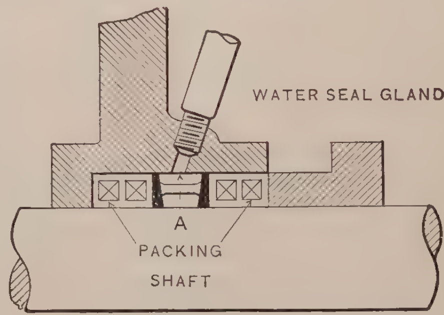


Fig. 12. Shaft stuffing box of Centrifugal Pumps, showing water seal.

72. **Packing.**—Use Worthington square graphite packing, or similar, for either hot or cold water service. Do not use flax packing on centrifugal pumps having bronze sleeves. Place one or two packing rings into bottom of stuffing box, then place water sealing gland so as to allow the cage to come to central position under water pipe, as shown on Fig. 12 at A. Next put in enough packing so that the gland may be drawn up loosely, allowing slight seepage along the shaft.

73. The **stuffing boxes** are **water sealed** to prevent air from leaking in and also to keep the packing wet. For this purpose a composition cage made in halves is inserted in the middle of the packing space and is supplied with water through a pipe connected to the stuffing box. If the pump handles clear, cold water, the supply for the water seal may be taken directly from the discharge. If the discharge pressure is less than 15 lb. per sq. in., or if the water handled contains sand and grit, or is hot, then an independent clean cold water supply at a pressure of 15 to 30 lb. per sq. in. must be used.

74. **Priming.—With a foot valve** installed on the suction pipe the pump can be primed by opening a vent cock or valve at the high point of casing and admitting water from some outside source until the suction pipe and pump casing are completely filled with water. Care must be taken to exhaust all air from suction pipe and pump casing as any air that remains entrapped will interfere or prevent the pump from lifting its water.

75. If a foot valve has not been installed on the suction pipe an **ejector** of some good design and of ample size should be connected to the high point on the pump casing. Ejectors can be operated with either air, steam or water. If water is used the supply must be of sufficient pressure to give a high velocity through the ejector.

76. If it is desired to make use of a **dry-vacuum pump for priming** purposes, care must be taken to protect the vacuum pump from getting any water.

77. Starting.—Before starting up **the first time,** try out the prime mover for correct direction of rotation with the coupling bolts removed from the coupling. The brass arrow on the pump casing shows the correct rotation.

78. Wash out all **bearings** thoroughly with kerosene and make sure the oil rings are free to turn. Fill the bearings with a good grade of dynamo oil. Check the **jacket cooling-water** piping on thrust bearings to make sure the inlet enters at bottom and discharges from top and then turn on the supply, allowing an ample amount of water to flow to keep the bearing cool. The water-seal supply to glands may be turned on and the pump primed. Do not run the pump unless primed and full of water, as there is danger of injuring some of the interior parts of the pump which depend on water for lubrication.

79. Final inspection should be carefully made of all parts of the unit before starting up, to make sure every part is ready for operation. When the pump is primed, the speed should be applied rapidly so that head is quickly generated to expel air that is entrained and to get a stream of water moving that will not break in continuity. If the stream breaks, the impeller will simply throw the remaining liquid out, and separate more air, and the pump will not start delivery. After a pressure is built up on the discharge, **open the discharge gate valve slowly.**

80. A **compound gage** connected to the suction of the pump and a pressure gage connected to the discharge and mounted at a convenient place will be a great help to the operator.

81. Stopping.—Always close the discharge valve before shutting down the pump, unless a check valve has been installed in the discharge line to prevent the water in the discharge line from flowing back and driving the pump backwards or exerting a high pressure on the foot valve.

82. Care and Operation.—During the routine operation of pumps the **bearings** should be occasionally **inspected** to determine whether or not there is sufficient oil in the oil pockets and if the oil rings are turning freely and supplying enough oil to the shaft and the bearings. The bearings should be drained, washed out, and the oil renewed every week or ten days during the first month of operation and as often after that as the conditions warrant.

83. The **stuffing-box glands** must be so **adjusted** as to permit a slight seepage of water out of the stuffing box at all times during operation, otherwise the packing will cause excessive wear on the shaft sleeves, as the stuffing boxes are dependent upon water for lubrication as well as for sealing.

84. Boiler feed pumps require careful **attention.** To avoid trouble the operator should see that no more pumps are running than are actually required. **Parallel operation** especially at one third of normal capacity or less is almost impossible with present-day governing devices, so that the operator must watch the load requirements and on the lighter loads close down all unnecessary units. **Do not run pump dry,** as the interior parts are dependent upon water for lubrication. Don't operate for any length of time with a closed discharge.

85. Do not use flax packing in stuffing boxes of pumps having bronze sleeves. Only in pumps having steel sleeves can flax or plastic packing be used.

86. The **dismantling of pump** should be done only when necessary. If it is found necessary at any time to renew the gaskets, use only gaskets of the original thickness. One side of the gasket should be painted with shellac and on the other side use a mixture of oil and graphite. Use particular care in getting all parts back to their original position when reassembling.

87. Locating Trouble.—If the pump does not operate satisfactorily the trouble may be due to one or more of the following causes:

88. Misalignment, distortion due to pipe strains, bearings badly worn, foundation not sufficiently rigid.

89. Low capacity may be due to: Clogged strainer; choked up impellers; below proper speed; excessive discharge head; air leaks; worn impeller; worn impeller rings.

90. Failure to pump may be due to: Failure to fully prime; air leaks; insufficient supply of water; wrong direction of rotation; below speed; excessive discharge head; excessive wear on impeller and impeller rings; clogged up strainer; stuck foot valve; air pockets in suction line; excessive suction lift; stuffing boxes not water sealed.

91. Ordering Repairs.—In ordering repair parts always give the pump serial number, which will be found stamped on the name plate on the pump casing. State the name and number of each repair part, for the particular class of pump the repairs are ordered for. If it is necessary to order any parts not shown clearly in the sectional cuts of the Repair Parts book, a sketch of the part should be included with the description.

INSTALLING STEAM-DRIVEN POSITIVE-DISPLACEMENT PUMPS

100. The **erection** of simplex or duplex steam-driven positive displacement pumps of the **smaller sizes** is a simple problem, as the steam and liquid ends have been properly aligned and rigidly bolted together before leaving the factory. The pump is placed upon a properly levelled foundation, the foundation bolts pulled up, steam and liquid connections made and the pump is ready to put in service. It is not necessary to align small pumps after they have been placed on the foundations. The **erection of large pumps** that are shipped in sections should be under the supervision of a competent erector, as it is necessary to carefully align the steam and liquid ends after they have been placed upon the foundations. If this is not done, excessive friction and wear will occur and the pump will not give satisfactory service.

101. Foundation bolts are used only on the liquid end of a steam pump. The steam end, owing to the linear expansion and contraction due to temperature changes, must be free and should not be bolted to the foundation. The steam end of large pumps should be placed on rollers to permit the free movement of the steam end in a longitudinal direction. These rollers are placed between the steam cylinder foot and a sole plate on the foundation.

102. Guide bolts are sometimes used to prevent any side movement of the steam end. These bolts are designed to allow the free longitudinal movement of the steam end.

103. A **feedwater heater** may be used to advantage **in the exhaust** line of large pumping engines. This heater is usually of the closed type and is located immediately back of the low-pressure cylinder. The general service and boiler-feed pumps in power plants should always exhaust into an open feedwater heater of the Worthington type.

104. The **steam cylinders** should be well **lubricated,** especially before starting, to prevent scoring the cylinders. On large pumps, hand-operated oil pumps should be provided on each steam cylinder for this purpose. The steam ends of small pumps may be fitted with a good sight-feed lubricator for cylinder lubrication. Large size pumps and pumps that operate with superheated steam should always be fitted with mechanical oil pumps for cylinder lubrication. The moving parts of the valve gear may be oiled by hand or by small oil cups properly located.

105. The **stuffing boxes** should be well and evenly filled with a good quality of **packing.** Do not screw glands up too tight or the rods and plungers will score. Use metallic packing in the steam end stuffing boxes when operating with superheated steam. Liquid pistons should be packed with a material that is suitable for the liquid being pumped.

106. Both the steam and liquid **cylinders** are provided with **drain** cocks. To prevent damage from freezing in cold weather, always drain the liquid cylinders. Always open the steam cylinder drain cocks before starting to allow condensate to escape and to prevent damage to cylinder heads.

SETTING THE STEAM SLIDE VALVES OF DUPLEX PUMPS

1.—*When moved by a single valve-rod nut working between lugs on valve (Fig. 13):*

107. First open drip cocks so that water in the steam cylinders will be completely drained away; then move the piston rod of one side toward the steam cylinder head by prying against the crosshead (not lever) until the steam piston strikes the head; make a mark on the rod close to the face of the steam-end stuffing-box follower, then move the piston rod to the

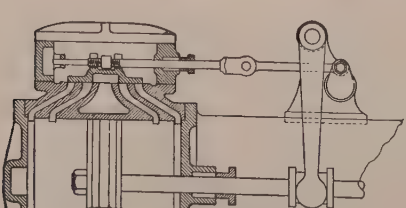


Fig. 13.

opposite end of the stroke until the steam piston strikes, and make a mark on the rod just half way between the first mark and the face of the steam-end stuffing-box follower. Now, move the piston rod backward until the second mark is flush with the face of the follower, and the piston will stand at mid-stroke. Disconnect the link from the knuckle of the valve rod on the opposite side and place the slide valve in the steam chest—the chest cover, of course, having been taken off for this purpose—so that the valve exactly covers both steam ports that lead to opposite ends of cylinder.

108. Now hold the slide-valve nut exactly in the center of the space between the slide-valve lugs; screw the valve rod through this nut until the knuckle eye is in line with the link eye and push the link pin in place. Repeat this process with the other side of the pump and the operation is complete. (It will be found an advisable plan to move both pistons to mid-stroke before touching either slide valve.) After everything is properly adjusted and before replacing the chest cover, be sure to move one of the slide valves off center so as to leave one steam port open, otherwise the pump cannot be started. In operation the valves can never move so that both will be on center at the same time under any condition of running. It is only when the valves are deliberately placed, as in the operation of setting, that it can happen.

2.—*When valve rod has lock nuts at each end of slide valve (Fig.* 14):

109. First, place pistons and slide valves on centers (See par. 107), but do not disconnect the valve rod from the link; then set and lock the nuts at equal distances from the outer faces of the valve lugs, allowing about half the width of the steam port for lost motion on each side. (A good way to prove equality of lost motion is to move the valve each way until it strikes the nut and note if both port openings are equal.) If it is found that this allowance gives the pump too much or too little length of stroke, the lost motion will have to be altered by trial until the pump makes the desired stroke.

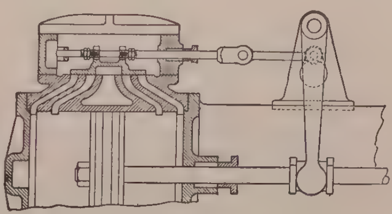


FIG. 14.

Too much lost motion lengthens the stroke and may cause the pistons to strike the cylinder heads; too little lost motion shortens the stroke, decreases the pumping capacity and increases the steam waste.

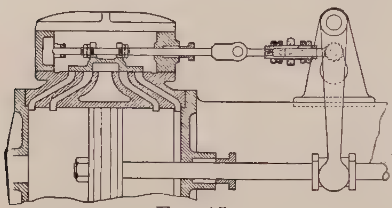


Fig. 15.

3.—*When slide valve has outside adjustment (Fig. 15):*

110. Set the piston in the middle of its stroke (See par. 107), likewise the steam valve on the opposite side; move the collars on the valve-rod link so that they will be about half the width of the steam port away from the tappet. Repeat this operation on the opposite side and the valves are set. Pump may now be started and if it is found that the stroke is too short, the collars must be screwed farther apart, care being taken to turn back all collars the same amount, for otherwise the piston rod movement will be less than full stroke and nearer to one end of cylinder than other. If the stroke is too long so that the pistons strike heads, the collars must be set closer together. With this type of valve gear it is not actually necessary to stop the pump and remove the chest cover, as the collars can be adjusted by trial until the required length of stroke is reached, while the pump is in regular operation. When the adjustment is finally made, be sure to lock the collars securely in place.

4.—*When valve is piston type with outside adjustment (Fig. 16): The cut shows a pump of vertical type.*

111. Set the main steam piston at mid-stroke, (See par. 107) likewise the steam valve on the op-

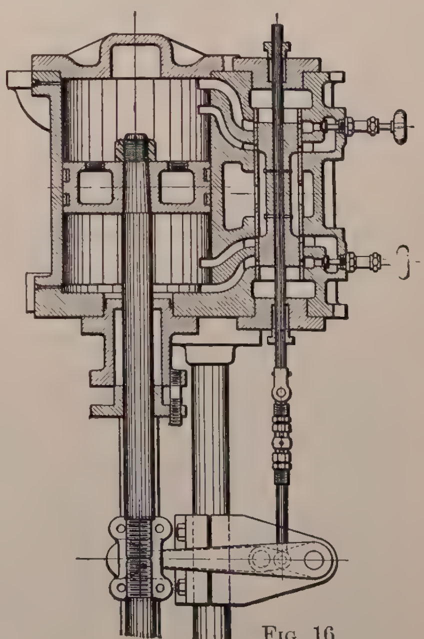


Fig. 16.

posite side. To set the valve in mid position, remove the top valve-chest cover and move the valve until its upper face is just in line with the top edge of the uppermost steam port. If the valve cylinder is of so small a diameter that it is difficult to set the valve on the port line by inspection, measure the distance from the upper edge of the top port to the upper face of the valve chest and move the valve until its upper end is the same distance below the face of the chest. When thus set, the lower end will be in line with the lower edge of the lower steam port as the valve is made without lap.

112. Friction between the valve and the cylinder bore or stuffing-box packing is usually sufficient to hold the valve in any position it may be placed, but if this is not the case the valve must be prevented from dropping, while adjustments are being made, by means of a temporary blocking of wood or whatever material may be available. After the main piston and steam valve are arranged in central position, the collars are set in the same manner as for an ordinary duplex slide valve with the outside adjustment, and, as is the case with these pumps having valves with outside adjustment, the valve travel and the length of stroke can be adjusted by trial while the pump is running, without the necessity of removing the chest cover.

5.—To set slide valves of a compound duplex pump:

113. Fig. 17 is a sectional view of a compound duplex pump, from which it will be seen that the valve mechanism is similar to that of a

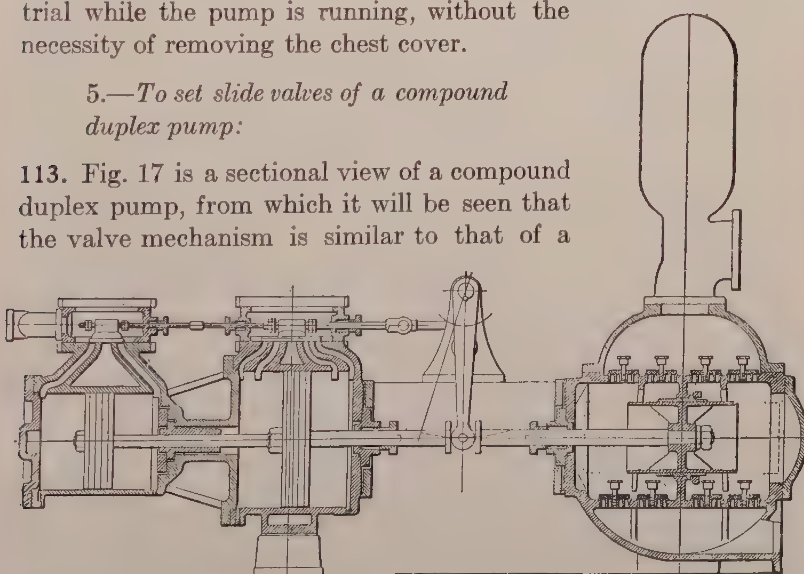


Fig. 17.

simple duplex pump, the difference being simply in the addition of another slide valve. These valves are set in exactly the same manner as for simple pumps, except that two valves are to be considered instead of one.

114. All pumps built at the Worthington Works have the steam valves properly set and are in perfect running condition at time of shipment.

115. If a pump does not operate satisfactorily do not touch the steam end until investigation shows that the trouble is not elsewhere. Most pump troubles are due, not to steam-end conditions, but to poor packing, fouled water cylinders, worn water valves or to faulty conditions in water pipe connections outside the pump itself.

116. In **designating** the two sides of a duplex pump, manufacturers have made it a practice to denote as **"right-hand"** that side of the pump at the right which is seen when standing at steam end and looking towards water end, the other, of course, being **"left-hand."** This applies to all horizontal pumps. In the case of vertical pumps, "right-hand" side is one seen when standing at and facing front side of pump as it stands vertically, and as steam cylinders of a vertical pump are always above water cylinders, right-hand side of a vertical pump is same as left-hand side of a horizontal pump. It is important to note this distinction in ordering repair parts.

INSTALLING POWER-DRIVEN POSITIVE-DISPLACEMENT PUMPS—DUPLEX AND TRIPLEX TYPES

120. If the **pump** is **direct-connected** by a coupling to its prime mover, care must be taken to have the driving shaft bearings of the pump in perfect **alignment** with the driving shaft bearings of the prime mover. The levelling of the pump and prime mover on the foundations and the alignment of the coupling between the pump and prime mover can be done in the same manner as described for centrifugal pumps.

121. If the **pump** is **connected** to its prime mover **by gears,** care must be taken not to mesh the pinion into the gear as closely as possible; but, instead, be sure to allow a slight amount of clearance or backlash. For small steel gears, approximately 0.002 to 0.003 in.

clearance will be sufficient. For large steel gears, the clearance should not exceed 0.005 to 0.006 in. If the driver pinion is of rawhide or similar material, the clearance or backlash should slightly exceed the figures stated, especially if considerable moisture is present, as a slight expansion may occur. The minimum clearance for a rawhide pinion should be approximately 0.003 to 0.004 in.

122. The **gears** between the pump and its prime mover must have a perfect bearing across the entire width of the face of the gear and pinion. To **test** this **tooth bearing** insert a piece of thin paper at each end of one tooth on the gear. Insert these pieces of paper on the bearing side of the tooth of the gear, then turn gear or pinion sufficiently to bring into mesh on the papered tooth. If the bearing across the face of gear and pinion is correct, the pressure on the two pieces of paper will be equal.

123. Both the pump and its prime mover must turn over freely both before and after the connection is made between the pump and prime mover. Flexible couplings are not intended to compensate for avoidable errors in alignment. Alignment should be practically perfect; and this is still more imperative when a cut-off coupling is used instead of a flanged coupling.

124. Every power-driven positive displacement pump should be equipped with a manually controlled **by-pass valve for use in starting,** especially where the pump is operated by an induction motor or other quick-starting type of driving agent. The use of a by-pass valve relieves the pump and piping of the shock and overload due to the inertia of the liquid column, and allows driver to start without the full load.

125. Do not allow a pump to run with **loose bearings.** This will destroy the bearings, and may in time cause cracks or breakage in the pump frame.

126. **Crosshead** shoes are tapered in thickness and provided with adjusting studs. These shoes should be **adjusted** when necessary to compensate for wear in crossheads or guides.

127. Keep all **bearings** constantly lubricated with as good quality of grease or oil as the lubricating attachments demand. Before **starting** for the **first time,** it is advisable to thoroughly clean out

all bearings and guides by pouring kerosene through them, as dirt and grit are likely to get in during transportation or installation.

128. The face of **gear and pinion** (herringbone or spur or any other type) must be kept thoroughly **lubricated**; a heavy thick graphite grease is recommended. Grease should not be used on rawhide; graphite and tallow may be used.

129. Gears operating **in oil-tight guards** or cases *must not be flooded* with oil; they should only *dip* in the oil. If flooded, there is a pumping action, and this heats the oil.

130. Keep the **stuffing boxes** well and evenly filled with a good quality of **packing.** Do not screw them too tight; and adjust them occasionally.

SECTION VII

Pumping in the Petroleum Industry

(Figures refer to paragraph number)

SECTION VII

Pumping in the Petroleum Industry

1. The word **Petroleum is derived** from two Latin terms "petra," rock and "oleum," oil. Rock oil, which was an early name given it in America, is accounted for by the fact that certain shales and coals possess oil as a part of their constituents. Petroleum is one of the family of bitumens which in their natural state assume many forms and are of world-wide distribution.

2. The occurrence of petroleum has been recorded from the earliest times. The first probable exploitation of petroleum in the way of distillation was by James Young in 1850.

3. The **original use of petroleum** was in the preparation of illuminating oil. After the production of illuminating oil from petroleum it was soon shown that the heavy petroleum oil possessed lubricating properties superior to the oils from vegetable and animal fats. At the present time practically all lubricating oils are obtained from petroleum.

4. Gasoline is now the most valuable product of petroleum. **Gasoline** was **originally used** for lighting purposes and for domestic stoves. The production of gasoline, however, was greatly in excess of the demand for these purposes. The development of the gasoline engine was due principally to the need of a commercial outlet for this surplus gasoline. With the development of the gasoline engine came the automobile, airplane and motor boat, all of which have increased the demand for gasoline to such an extent that refineries are today spending large sums of money for apparatus to extract the greatest possible amount of gasoline from the crude petroleum.

5. **Crude petroleums** from different parts of the world and even from different formations in the same locality differ widely in their properties and composition. In some cases the oil is found almost white. From this it varies through all the shades of amber and brown to black. It is found as highly liquid as water and again with a viscosity such that it will hardly flow.

6. The manufactured products derived from the different grades of crudes have different properties as well. From some crudes special lubricating oils can be made which cannot be manufactured from other grades of crudes, and some lamp oils possess greater illuminating power than those derived from other crudes. This is not due to the methods of refining but to the actual difference in the properties of the refined oils derived from different crudes.

7. As petroleum and most of its products are liquids or semi-liquids, its economical handling is largely a pumping problem. Generally speaking, it is pumped from the well through the various stages of transportation and refining to the point of ultimate use.

8. The **pumping requirements of the oil industry** may be divided into three general classes, as follows:

1. Well pumping
2. Transportation, crude and refined oils
    - Pipe line
        - Feeder
        - Trunk
    - Loading to
        - Tankships and barges
        - Tank cars
    - Cargo unloading to shore from
        - Tankships
        - Barges
    - Fuel Oil
        - To Tanks on Shore
        - To Burners on Ships
3. Refining; throughout the entire process.

9. The **pumping** of petroleum and its products presents a more difficult **problem** than the pumping of water. The paraffin-base crudes are of low viscosity, and flow freely. The asphalt-base crudes are of high viscosity, thick, sluggish, and require preheating to lower the viscosity and render them sufficiently fluid to flow with any degree of freedom. The refined products, such as gasoline, naphtha, and kerosene, are as fluid as water, but they are more difficult to handle than water. The total absence of lubricating qualities in gasoline creates a serious problem in packing liquid pistons and piston-rod or shaft-stuffing boxes. The solvent action of naphtha on certain metals, and the tendency of kerosene to seep through the most minute openings must be considered and provided for. These problems are not only troublesome to the pump manufacturer but to the pump user as well.

10. There are two properties that largely determine the mechanical behavior of liquids. These are **specific gravity** (density) and **viscosity** (relative fluidity). The specific gravity is the relation by weight of the same volume of oil and water.

$$\text{Spec. Gravity} = \frac{\text{Weight Oil}}{\text{Weight Water}}$$

Unless some other temperature is specifically mentioned, the gravity refers to 60 deg. F. In tables the specific gravity is given at 60 deg./60 deg. F. which is the specific gravity of the oil at 60 deg. F. as compared to water at 60 deg. F. as unity.

11. The absolute specific-gravity scale is not commonly used in the oil industry, as all the readings are in decimals. Instead, an entirely arbitrary standard scale, called the **Baumé Scale,** is used. For the Baumé scale the gravity of water is taken as 10 deg. Liquids heavier than water are designated in degrees below 10, and liquids lighter than water are designated in degrees above 10. Two Baumé gravity scales are in use in the oil industry. One is that adopted by the U. S. Bureau of Standards, and its relation to specific gravity is expressed, for liquids lighter than water, by the formula:

$$\text{Spec. Gravity } 60°/60° = \frac{140}{130 + \text{Degrees Baumé}}$$

$$\text{and Degrees Baumé} = \frac{140}{\text{Spec. Gravity } 60°/60° \text{ F.}} - 130$$

For liquids heavier than water:

$$\text{Spec. Gravity } 60°/60° = \frac{145}{145 - \text{Degrees Baumé}}$$

$$\text{and Degrees Baumé} = 145 - \frac{145}{\text{Spec. Gravity } 60°/60° \text{ F.}}$$

12. Another scale in common use is that of the American Petroleum Institute, which is based on the following relation to specific gravity:

$$\text{Spec. Gravity} = \frac{141.5}{131.5 + \text{Degrees Baumé}}$$

$$\text{and Degrees Baumé} = \frac{141.5}{\text{Spec. Gravity } 60°/60°} - 131.5$$

for liquids lighter than water.

13. The difference in the readings of the Bureau of Standards and the A.P.I. Baumé scales varies from 0 with very heavy oils to as much as 0.5 deg. Bé. for light distillates such as gasoline.

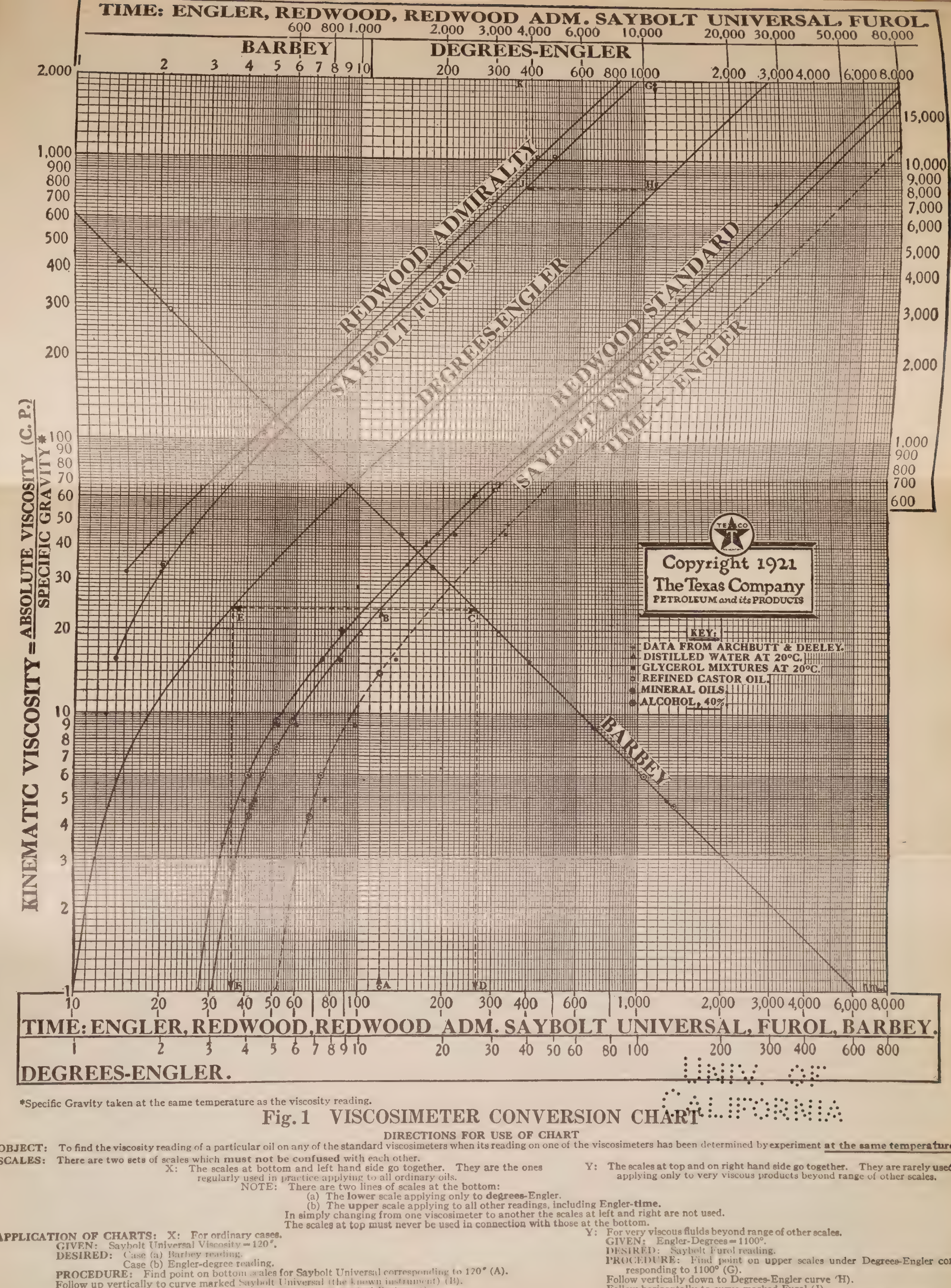
14. The U. S. Bureau of Standards, the U. S. Bureau of Mines and the American Petroleum Institute have agreed that in the interest of uniformity of practice the modulus 141.5 be used exclusively by the oil industry, and be known as the **"American Petroleum Institute" Scale.** Data are to be expressed as "Degrees A.P.I." to avoid confusion with the 140-modulus Baumé scale. The U. S. Bureau of Standards requests that the designation "Degrees Baumé," when referring to liquids lighter than water, be used only when the modulus 140 is the basis of calculations.

15. Viscosity is the resistance which the particles of a liquid offer to sliding past each other. Low viscosity means that the particles slide easily, as in water, gasoline and thin, light oils. High viscosity means that the particles do not slide easily, as in tar and asphalt-base crude oils.

16. The **viscosity** of an oil is the **measure** of its fluidity, and it is expressed in seconds. The number of seconds indicates the time required for a given amount of oil at a known temperatures to flow through a small orifice. There are a number of **viscosimeters** in use today, so "viscosity" conveys no meaning unless the name of the instrument and the temperature of the oil tested are given.

17. In the United States the **Saybolt Standard** Universal Viscosimeter is most generally used. The **Redwood Viscosimeter** is used extensively in Great Britain. In Germany and other countries in Europe the **Engler Viscosimeter** is used.

18. The **Barbey Viscosimeter** measures the quantity of flow per hour under constant head instead of the time for a definite quantity under varying head, as do the others. The **Saybolt Furol** and **Redwood Admiralty** are especially designed for measuring the more viscous oils, and they give a reading in seconds approximately one-tenth of that given for the Saybolt Universal and Redwood, respectively. Curves for the conversion of readings on the Saybolt, Redwood, and Engler viscosimeters are given in Fig. 1.



*Specific Gravity taken at the same temperature as the viscosity reading.

Fig. 1 VISCOSIMETER CONVERSION CHART

DIRECTIONS FOR USE OF CHART

OBJECT: To find the viscosity reading of a particular oil on any of the standard viscosimeters when its reading on one of the viscosimeters has been determined by experiment **at the same temperature.**

SCALES: There are two sets of scales which **must not be confused** with each other.

X: The scales at bottom and left hand side go together. They are the ones regularly used in practice applying to all ordinary oils.

Y: The scales at top and on right hand side go together. They are **rarely used,** applying only to very viscous products beyond range of other scales.

NOTE: There are two lines of scales at the bottom:
(a) The **lower** scale applying only to **degrees**-Engler.
(b) The **upper** scale applying to all other readings, including Engler-**time.**
In simply changing from one viscosimeter to another the scales at left and right are not used.
The scales at top must never be used in connection with those at the bottom.

APPLICATION OF CHARTS: X: For ordinary cases.
GIVEN: Saybolt Universal Viscosity = 120".
DESIRED: Case (a) Barbey reading.
Case (b) Engler-degree reading.
PROCEDURE: Find point on bottom scales for Saybolt Universal corresponding to 120" (A).
Follow up vertically to curve marked Saybolt Universal (the known instrument) (B).
(a) To get Barbey reading: Follow horizontally to curve marked Barbey (C).
Drop down to bottom scale for Barbey and read (C). Reading = 265°.
(b) To get Engler-Degree reading: Follow horizontally to curve marked Degrees-Engler (E).
Drop down to lower of the two bottom scales, for Engler-Degrees and read (F). Reading = 3.62°.
i. e., 120" Saybolt = Barbey 265° = Engler 3.62.

Y: For very viscous fluids beyond range of other scales.
GIVEN: Engler-Degrees = 1100°.
DESIRED: Saybolt Furol reading.
PROCEDURE: Find point on upper scales under Degrees-Engler corresponding to 1100° (G).
Follow vertically down to Degrees-Engler curve (H).
Follow horizontally to curve marked Furol (J).
Rise vertically to scales under Furol and read (K). Reading = 3800".
i. e., 1100° Engler = 3800" Furol.

19. The **Standard Saybolt Viscosimeter** is made entirely of metal. The standard oil tube is fitted at the top with an overflow cup and the tube is surrounded by a bath. At the bottom of the standard oil tube is a small outlet tube through which the oil to be tested flows into a receiving flask. Thermometers are provided to show the temperatures in the bath and in the standard oil tube.

20. The time in seconds for the delivery of 60 cu. cm. of oil is the Saybolt viscosity of the oil at the **temperature at which the test was made.** Viscosity is commonly determined at 100 deg. F., 150 deg. F. or 210 deg. F.

21. Light oils, gas oils, straw, engine, dynamo, auto, cottonseed and similar oils may be tested at 70 deg. F. **Engine oils,** machine oils, and, occasionally, cylinder oils may be tested at 100 deg. F. **Heavy oils** such as cylinder and road oils and asphaltic fluxes may be tested at 210 deg. F. **Asphalt,** fluxes, paraffin wax and residues may be tested at 338 deg. F.

22. The **viscosity of fuel oils and road oils** is determined by means of the **Saybolt-Furol Viscosimeter.** The apparatus and method of test are the same as for the Saybolt Universal. All dimensions of the instrument are the same except the inside diameter of the outlet tube which shall be, minimum 0.313; normal 0.315; maximum 0.317 cm. Viscosity may be determined at 70 deg. F.; 104 deg. F.; 122 deg. F., and is expressed as Saybolt-Furol.

23. Redwood, Engler, or Barbey viscosimeters are not used in the United States, and will not be discussed here. A chart for converting Saybolt to Redwood or to Engler or vice versa will be found in Fig. 1.

24. A chart for converting Saybolt Universal to Saybolt Furol is given in Fig. 1.

25. The **viscosity** of oil decreases and the **fluidity** increases very much when **oil is heated.** Full advantage is taken of this when certain kinds of crudes and fuel oils are to be pumped. When steam pumps are used the exhaust steam is utilized for heating the oil. When power pumps are used the oil may be heated by steam from boilers installed for that purpose. The effect of temperature on viscosity is shown by table, par. 26 and chart Fig. 2.

26. CHARACTERISTICS OF DIFFERENT OILS

Kind of Oil and Source	Gravity at 60° F.		Viscosity at Different Temperatures ° F.									
			Saybolt Universal					Absolute in Centipoises				
	Spec.	Bé.	32°	40°	60°	80°	100°	32°	40°	60°	80°	100°
Ozark Pipe Line	0.8103	43.1	48.	44.	40.	38.	36.	5.55	4.65	3.38	2.70	2.24
(Cushing, Okla.)	0.8586	33.05	264.	200.	125.	93.	77.	48.	36.6	22.	15.6	11.9
Stroud (Okla.)	0.8156	42.	56.	50.	43.	39.	37.	7.24	5.92	4.04	3.10	2.50
Ranger (Texas)	0.8260	39.5	100.	76.	53.	43.	39.	16.45	11.8	6.56	4.3	3.1
West Columbia (Texas)	0.9162	22.8	C	3215.	783.	381.	220.	C	650.	155.	74.5	42.1
Sour Lake (Texas)	0.9239	21.6	C	2240.	721.	365.	207.	C	450.	142.	69.7	40.
Salt Creek (Wyoming)	0.861	33.9	3540.	1280.	219.	71.	42.5	665.	240.	40.	11.	4.15
Lost Soldier (Wyoming)	0.8775	30.	C	C	1215.	165.	49.	C	C	230.	30.	5.95
Santa Fe Springs (Cal.)	0.8605	32.7	243.	165.	82.	59.	47.	45.1	30.	13.1	8.25	5.28
Signal Hill (Cal.)	0.9025	25.7	4480.	1225.	495.	180.	90.	880.	240.	94.	34.	1.55
Colinga (Cal.)	0.919	22.8	.....	.....	4840.	175.5	80.5	.....	.....	960.	34.5	1.55
Penna. Crude	0.807	49.4	52.8	50.5	46.	42.8	41.	6.75	5.95	4.74	4.02	3.53
			50°	100°	200°	300°		50°	100°	200°	300°	
Mexican Crude	0.975	13.5	417,000.	14,800	536.	94.		90,000.	3,000.	107.	18.2	
Fuel Oil (Okla.)	0.938	19.	1,810.	775	...	185.		391.	150.	...	32.5	
			50°	100°	300°	600°		50°	100°	300°	600°	
Gas Oil	0.812	43.5	220.	32.8	29.	28.9		39.	1.28	0.065	0.0027	
Diesel Oil (Cal.)	0.854	34.6	58.	37.6	30.5	29.5		8.2	2.82	0.53	0.185	

C—Oil congeals at this temperature.

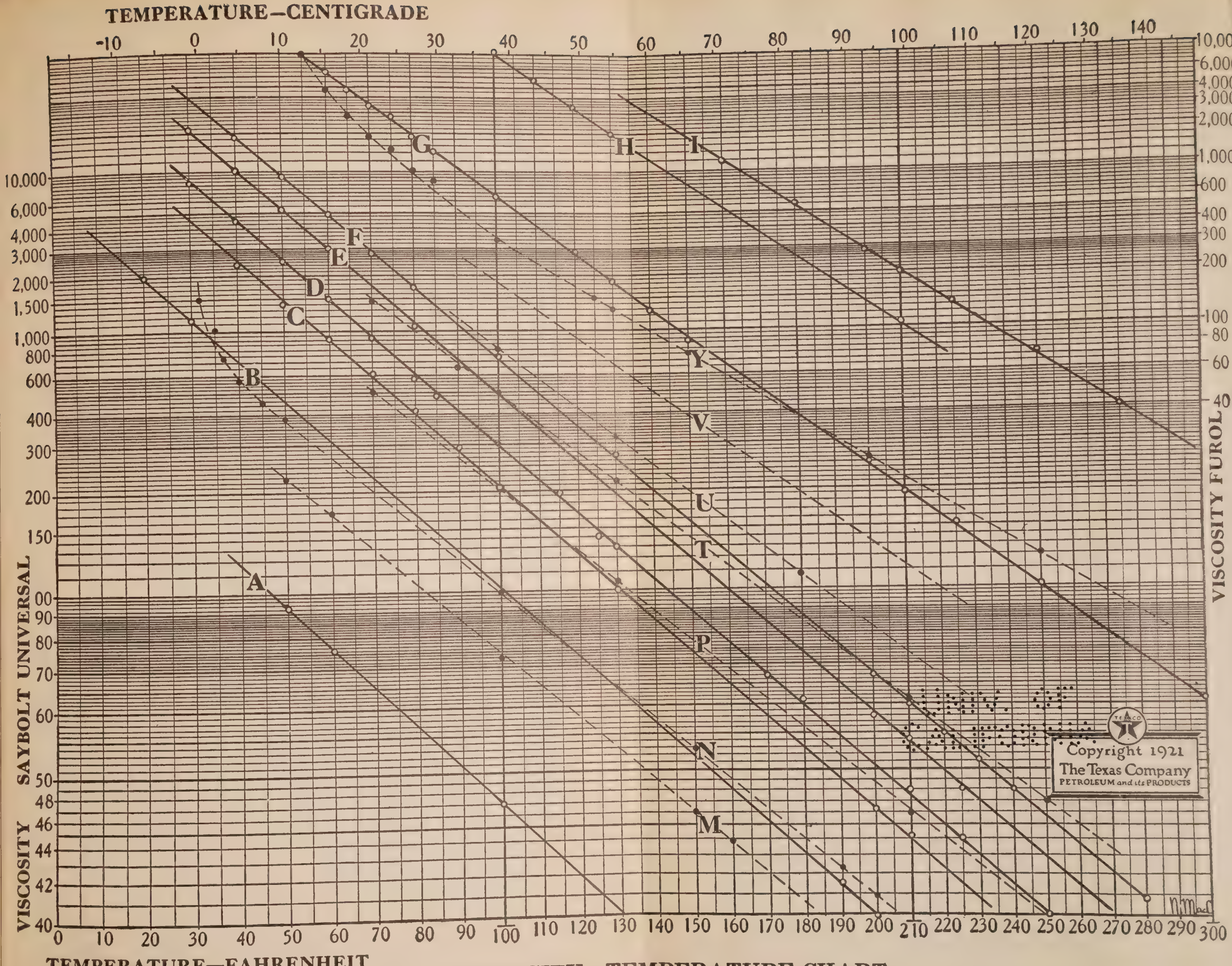


Fig. 2 VISCOSITY—TEMPERATURE CHART

27. It is evident from a little study of the properties of oils, especially the crudes, that **pumps for handling oil** must be carefully designed. The low viscosity of water permits a high velocity and small liquid passages, which in many designs are long and tortuous. The high viscosity of crude and fuel oils requires the utmost skill in designing the liquid ends of pumps. The liquid passages must be short, direct, and free from abrupt turns. Bends or turns, where necessary, must be of easy, graceful flow lines. The liquid valves must be designed for simplicity and maximum area for space occupied and must be so located as to give straight-line flow.

28. So far as possible, **parts** should be **interchangeable.** The use of right and left-hand pieces should be avoided in order to reduce the number of spare parts to be carried in stock.

29. It is very necessary that **parts** for pumps used in the oil industry be quickly and easily **replaced** and parts for replacement be carried in the field stock. Many pumping stations are located in out-of-the-way places and often with very poor railroad connections. On a pipe line, each station forms one link in a chain many hundreds of miles long, and its failure to operate will tie up the system from end to end, causing great losses, both direct and indirect. In a refinery, pump failures may curtail production at a critical time. Hence the necessity for (1) so designing and building the pumps as to render them as nearly immune from breakdown as is humanly possible; (2) carrying on hand either complete spare units, or a complete assortment of those parts most liable to break or to wear out; and (3) entrusting the care and operation of the pumps only to skilled, careful and conscientious men.

30. Drilling for oil is a tedious and frequently a long operation. The usual method of drilling is with a heavy string of tools hung on a rope or cable passing over a pulley located at the top of a derrick 60 to 100 feet high. A string of tools is about 40 feet long and consists of a bit or drill, auger stem, jars, sinker and rope or cable socket. When attached to the cable they are suspended in the derrick and lowered into the well. The drilling engine is started and the speed adjusted. The tools are then fed out so that the bit strikes an effective blow. When the bit shows signs of not falling freely the tools are raised to the surface, the bailer is lowered and the well cleaned out, after which drilling is resumed.

Cable drilling is always used for prospect wells, as the material brought to the surface by the bailer enables the operator to know the character of the formation through which the drill is passing, which is very important in prospecting work.

31. Rotary drills are successfully used on a large scale in established fields where the producing horizon is definitely known.

32. The rotary motion is transmitted by means of a pipe to a special bit. The circulating fluid for removing the cuttings is pumped over the bit under pressures of 150 pounds per square inch and upward, by **"slush pumps."** **Cementing of wells** is done with Worthington Piston-pattern Pressure Pumps. See table, par. 155 of this section.

33. The production of oil when there is no natural flow or when the natural flow has stopped is obtained by the use of ordinary lifting pumps. For **pumping wells,** the Worthington "Glendora" Pump is suitable when the level of the oil is not more than 500 feet below the surface of the ground.

34. In the early days of the oil business, five-gallon cans, and pack horses or mules, constituted the "tank cars" of the oil trade. As the industry developed, barrels and tank wagons replaced the pack horses. In Pennsylvania some of the oil was transported by barges fitted with wooden tanks. Today the producer puts a tank at his well, into which the crude oil flows or is pumped. From this tank the oil is pumped through pipes by pumps to the tank farm belonging to the transporting company.

35. From the tank farm in the oil fields, the crude oil must frequently be transported a considerable distance to the refinery. The most economical method of transporting the oil is by means of **pumping through pipe lines.**

36. The first pipe line for transporting petroleum was laid in 1862. This was a 2-in. line about $2\frac{1}{2}$ miles long from the Tarr Farm to the Humboldt Refinery at Plumer, Pa. The growth of pipe lines was slow at first, lines being laid only from the wells to local refineries. The first trunk line was laid in 1875. This was a 6-in. line from the lower field of Butler Co., Pa., to Pittsburgh, Pa., a distance

of about 40 miles. In 1880 a trunk line of over 100 miles in length was laid from Butler Co., Pa., to Cleveland, Ohio. The next trunk lines were laid from Bradford, Pa., to the Atlantic seaboard. The modern pipe line is laid underground and paralleled by telephone and telegraph lines. Long trunk lines are usually of 6-in. or 8-in. pipes good for 1000 pounds pressure.

37. At intervals of 25 to 70 miles, depending on the oils, **pumping stations** are established with pumps capable of maintaining pressures of 750 to 1000 lb. per sq. in.

38. At each station are receiving and dispatching tanks connected with the pumps by manifolds, so that the receipt and dispatch can be switched from one tank to another, as required.

39. Pipe-line pumping is concerned with both the pumps and the pipe-line, and unless **both** are properly co-ordinated and designed to work together, good results cannot be expected. The **static load** is usually small, except in crossing mountains. The **frictional load** is dependent on the viscosity of the oil and on the velocity through the pipe line. With constant viscosity, the work of pumping varies theoretically with the cube of the speed. As all crude oils and even refined oils, except for the very light distillates, are vastly more viscous than water, we see at once how important become all such features of design and construction as angles, bends, valves, passage-ways in liquid ends, etc.

40. For pumping oil through pipe lines, so far as general types of pumps used are concerned, practically all that has been already stated with respect to reciprocating pumps will apply, bearing in mind that the pump must be especially designed with respect to liquid valves and passages.

41. Materials used for packing, gaskets, etc. must be of a nature that will not be affected by the oil. All joints and stuffing boxes must be kept in good order, as oil-leaks not only mean the loss of a valuable material, but are both dangerous and unsightly.

42. The **fire-risk** should be carefully guarded against, so far as possible; for while the oil itself may not be highly inflammable, the vapors given off are very much so, a very low percentage of such vapors or gases forming with the atmosphere a highly explosive

mixture. This makes **good ventilation around the pump-rooms** an important consideration, and prohibits smoking and the use of oil lamps or lanterns around the pumps.

43. For close inspection around dark places the pocket flash light is safe and convenient, while for general **illumination** ordinary incandescent electric lights, properly safeguarded, may be used. Ample **fire-fighting equipment** should be supplied and all employees drilled in its quick and efficient use.

44. The question as to the proper means of **driving** the **pumps** has already been discussed; in a word, whatever method is most economical under the local conditions. If steam power is available, the direct-acting duplex steam pump, compound or triple expansion, is a flexible, reliable, and fairly efficient machine, with a remarkably low upkeep and maintenance expense. Duplex and triplex power pumps are frequently used, and may be driven by any approved and convenient method.

45. Until recently the **calculations of the flow of oil in pipes** has been approached with considerable misgivings by engineers. Data were scattered, and none too concordant. In 1882 Reynolds conducted experiments in which he introduced coloring substances into water flowing through a glass tube, and these showed that at low velocities the flow was stream line, or viscous, and at high velocities the flow was sinuous or turbulent. In the **stream** line or viscous **flow** the liquid seems to pass through the pipe in a manner similar to the movement of a large number of concentric tubes, those at the center of the pipe having the greatest velocity, and those near the surface of the pipe the least velocity. Stream line or viscous flow is illustrated graphically by Fig. 3.

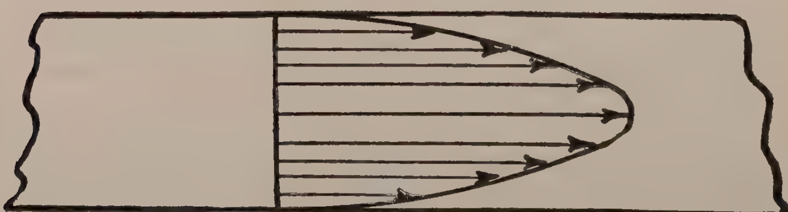


Fig. 3. Stream line or viscous flow.

46. A liquid in **turbulent** motion or **flow** acts more like a number of small spheres which roll over each other as they pass through the pipe and constantly change their relative positions both longitudinally and transversely, as shown graphically by Fig. 4.

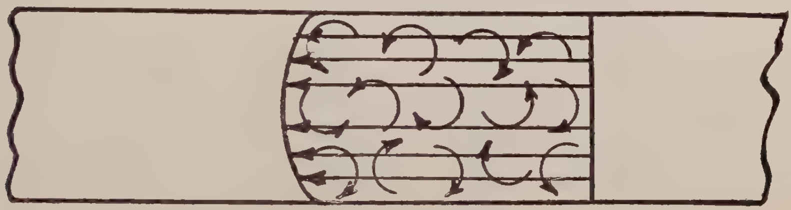


Fig. 4. Turbulent flow.

Whether a liquid is following stream-line or turbulent flow depends upon its velocity, its viscosity, the diameter of the pipe, and its internal condition. The point at which the flow passes from one condition to the other is known as the "critical velocity," the value of which has been found to decrease as the viscosity decreases. and as the pipe diameter increases.

47. Reliable data on the **friction of oil in pipes** is none too plentiful. Much investigation of this subject has been done by The Texas Company and we reprint by their permission the following very able discussion of the entire subject of the flow of oil in pipes, which appeared in their publication, *Lubrication.* This discussion considers both the case of stream-line or viscous flow, and turbulent flow.

48. If in flowing through a pipe the oil followed the **viscous-flow law**; that is, if the velocity was sufficiently low, so that the liquid flowed like a large number of concentric tubes without intermixing, a certain definite formula, **Poisseuille's equation,** held with fair accuracy. This equation simply said that the pressure necessary to force oil through a definite length of pipe was directly proportional to the velocity and viscosity of the liquid and inversely proportional to the area of the pipe cross-section opening.

49. When, however, the velocity reached a certain critical value or region, the liquid became turbulent and acted like a lot of balls

rolling over each other. In this case Poisseuille's equation did not hold, and several attempts had been made to formulate the pressures necessary but these did not agree very closely with each other.

50. "Some recent work carried on at the Massachusetts Institute of Technology, 'The Flow of Fluids through Commercial Pipe Lines,' by Robert E. Wilson, W. H. McAdams, and M. Seltzer, *Jour. of Industrial and Engineering Chemistry*, Vol. 14, No. 2, page 105, has greatly cleared these discrepancies and given us a method of **calculation** much simpler than any previous one. It was shown that one formula could be used to cover practically all cases, provided the constant was varied to a fairly definite law. This equation, originally developed by Fanning, is as follows:

$$p = \frac{Kflsv^2}{D}$$

where p is the pressure; f the variable friction factor; l the length in ft.; s the specific gravity of the liquid; v the velocity of the liquid; and D the diameter of the pipe. K is a constant depending on units used, and not on independent functions of all four variables, as has been generally assumed.

51. "It was shown by Blausius in 1912 that f was a function of $\frac{Dvs}{z}$ where z is the absolute viscosity of the liquid. In the case of viscous flow, the relationship followed the simple law

$$f = 0.00207 \frac{z}{Dvs},$$

which if substituted in Fanning's equation gave that of Poisseuille. In the region of turbulent flow, however, the relationship was not so simple, but it was definite; that is, for every value $\frac{Dvs}{z}$ there was a definite f. Therefore by calculating $\frac{Dvs}{z}$ for any particular case, f was determined; which, substituted in Fanning's equation, gave the correct value of the pressure drop.

52. "Messrs. Wilson, McAdams and Seltzer made a large number of determinations of f from their own data on several sizes of pipes and various viscosities of oils, and at different velocities. These values were plotted on a chart similar to Fig. 5. All the reliable

Formula for computing pressure necessary to force Fluids through Pipes.

$$p = \frac{0.323\, f\, l\, s\, v^2}{D} = \frac{0.0538\, f\, l\, s\, Q^2}{D^5}$$

p = pressure drop in lbs. per sq in. s = specific gravity referred to water

f = friction factor (Fanning) D = Inside pipe diameter in inches

l = length of pipe in feet Q = flow in gallons per minute

v = average linear velocity in pipe line (ft. per sec.) $= \frac{0.408Q}{D^2}$

z = absolute viscosity in centipoises = kinematic viscosity + specific gravity Kinematic viscosity may be determined from tables or approximately by formula $K = 0.216\,T - \frac{180}{T}$, where T = Saybolt time in seconds.

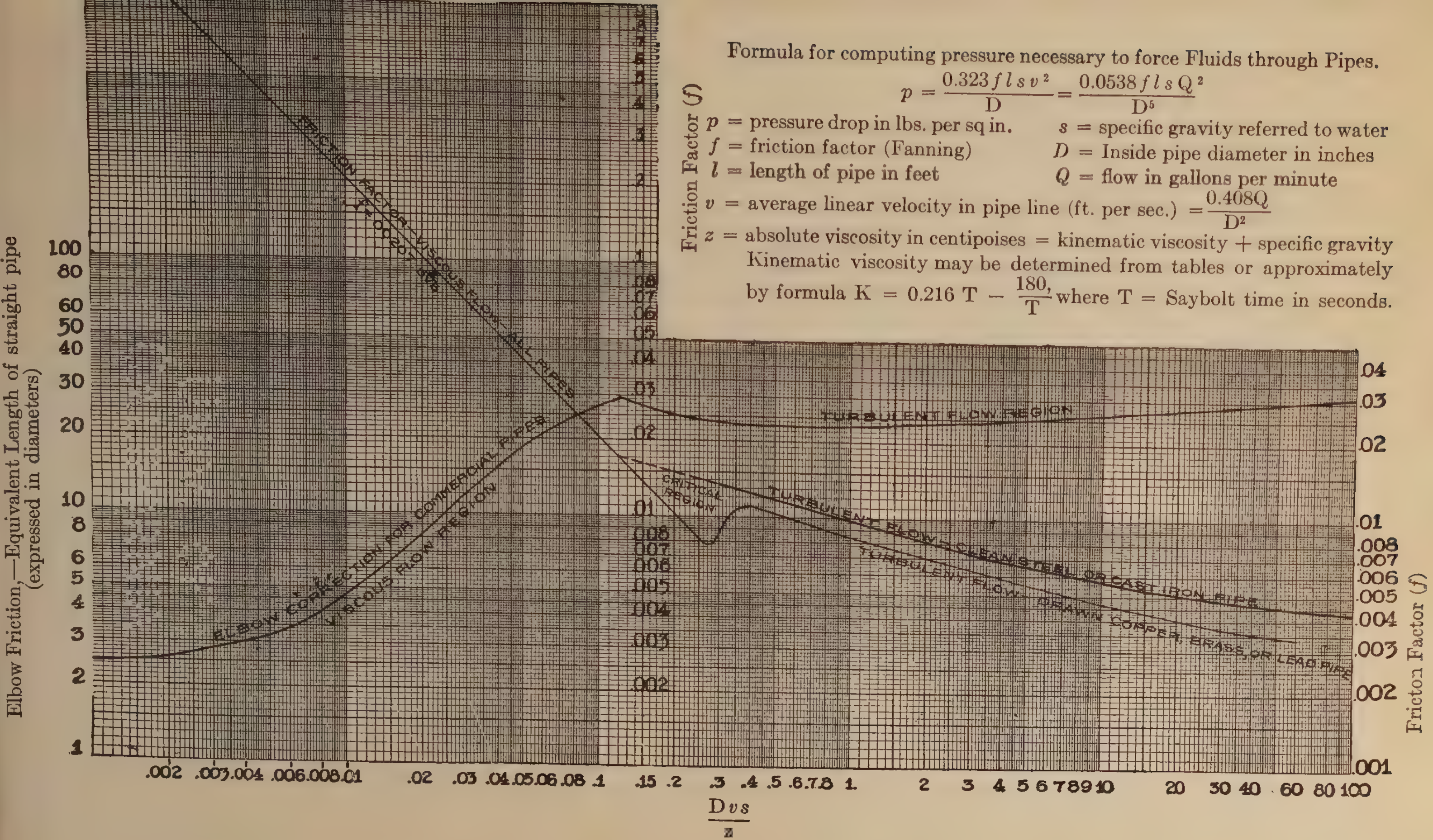


Fig. 5

Chart for Determining Friction Factors Used in Computing Fluid Flow in Pipes

Curves copied from charts furnished by Research Laboratory of Applied Chemistry, Massachusetts Institute of Technology

available data from other experiments on gases and liquids were plotted also, and found to agree reasonably well with their own experiments, so this can now be considered the latest word on flow of oil.

53. "It is to be noted in this plot that in the turbulent flow region the value of f varies according to the condition of the inner surface of the pipe. This was to be expected, but with a little judgment, an engineer should have no difficulty in getting true values.

54. "Messrs. Wilson, McAdams and Seltzer also determined the **effect of elbows** upon the pressure drop in a line of pipe. 'The most convenient method of expressing the frictional resistance due to an elbow of a given size is in terms of "equivalent length" of straight pipe which will give the same resistance to the flow of a given liquid.' It has been found practical to express this as a function of the modulus $\frac{Dvs}{z}$ and the average resulting curve from many experiments is shown in Fig. 5. The ordinate in this plot shows the number of pipe diameters which must be added to a length of pipe to allow for the effect of each elbow. This is computed against the value of $\frac{Dvs}{z}$, which an engineer will have computed already in determining f.

55. "The **calculation** of the **pressure drop through a pipe line** under any specified set of conditions is a very simple matter. The steps involved may be outlined as follows:

1. Calculate the value of $\frac{Dvs}{z}$ (or its equivalent, $0.408 \frac{Qs}{Dz}$ if Q* is known rather than v) for the fluid and pipe in question. If there is any uncertainty as to the measurement or estimation of any of the quantities, see the subsequent section on determination of the known quantities.
2. By referring to Fig. 5, find the value of f which corresponds to this value of $\frac{Dvs}{z}$ using the curve corresponding to the degree of roughness of the pipe in question. For values of $\frac{Dvs}{z}$ less than 0.002, extrapolations on the straight 45-deg.

*Q = gallons per minute. See par. 59.

line for viscous flow may readily be made by dividing the abscissae by 10 (or 100) and simultaneously multiplying the ordinate by 10 (or 100).

3. Using the same value of $\frac{Dvs}{z}$, refer to Fig. 5 for the equivalent length of straight pipe corresponding to each 90-deg. elbow. Multiply this by the number of elbows, convert the connection from pipe diameters into feet, and add it to the length of straight pipe. (Note that two elbows very close together, or a return bend, have less effect than two elbows some distance apart.)

4. Insert this value of f in the modified Fanning equation, $p = 0.323\frac{flsv^2}{D}$ or its equivalent, $p = 0.0538\frac{flsQ^2}{D}$, and obtain the pressure drop directly. This method and this formula should be used regardless of whether the liquid is in viscous or turbulent motion.

56. "The foregoing procedure must, of course, be slightly modified in case the permissible or available pressure drop is known and it is desired to calculate any one of the other five quantities which may be unknown in a given, practical problem. The factors used in the Fanning equation are all easily computed with little danger of error except perhaps the specific gravity (s) and the viscosity (z).

57. "The **value of the specific gravity** s, must be expressed in units relative to water at 39 deg. F. (temperature of maximum density). In other words, it is equivalent to the true density in grams per cubic centimeter. Specific gravity figures are frequently given relative to water at some other temperature than 39 deg. F. (4 deg. C.), and hence are subject to some slight corrections to give the proper value of s for use in the foregoing equations."

58. "The factor viscosity (z) in this article is the absolute **viscosity** and is **expressed in centipoises.** In order to obtain absolute viscosity both the Saybolt viscosity and the specific gravity must be known. There are a number of tables giving the relationship between Saybolt time and kinematic viscosity, which is absolute viscosity divided by specific gravity. One formula recently developed is

$$\text{Kinematic viscosity} = 0.216\,T - \frac{180}{T}$$

where T is the Saybolt time in seconds.

59. "**Velocity** should be expressed in feet per second, and means the effective average velocity in the pipe, obtained by dividing the rate of discharge in cubic feet per second by the cross-sectional area of the pipe in square feet. Data may be transposed from gallons per minute to velocity in feet per second, or vice versa, by the conversion formula,

$$v = \frac{0.408Q}{D^2}$$

(In this formula, D is inches, Q is gallons per minute and v is feet per second).

60. "The data and methods of calculation here given cover substantially the entire field of the flow of all fluids, including gases and dry vapors, through commercial pipe lines. There are, however, a few **exceptions** which should be carefully noted before attempting to use the formula in certain special cases. These may be specified as follows:

1. The method is not directly applicable to the **flow of gases** where the drop in pressure along the pipe line is more than 10 or 15 per cent of the final **absolute** pressure, because the density and velocity of the fluid are thereby changed. If, however, average values are used for the density and velocity, instead of the initial or final values, this formula may be used for pressure drops up to 40 or 50 per cent without serious error.

2. The method is not applicable to **flow through short sections of pipe** opening into large chambers where the entrance and exit losses are appreciable in comparison with the friction losses through the pipe. These losses become inappreciable where the pipe is more than 1000 diameters long, and may generally be neglected for approximate calculations for lines longer than 300 diameters.

3. The method is not applicable to the flow of semi-solid plastic materials, such as asphalt, clay solutions, very viscous colloidal solutions, etc., where the laws of flow are

modified by the tendency of the material to behave like a solid under certain conditions. However, the curves apply to certain **crude oils,** four thousand times as viscous as water at room temperature.

4. Precautions must be observed and highly accurate results cannot be expected in cases: (a) where the pipe is badly corroded or tuberculated, and the apparent value of f may increase to double that given in the formula, owing partly to a decrease in the effective cross-section area; (b) where a hot oil is passing through a pipe line with viscous-flow motion, and the existence of a large temperature gradient from the inside to the outside makes accurate calculations practically impossible; and (c) where there is any tendency to precipitate out solids, such as paraffin wax, on the walls or in the bottom of the pipe lines.

61. "With these exceptions, it is believed that the recommended method of calculation is applicable to an entirely satisfactory degree of accuracy to all commercial problems involving the flow of liquids through pipe lines."

62. The foregoing matter, and Fig. 5, should enable anyone with a little mathematical knowledge to figure out almost any oil-line pumping problem.

REFINERIES AND PROCESSES

70. The **purpose** of the **refining process** is to produce from that complicated mixture of compounds, crude petroleum, the various marketable products in which it abounds. This is accomplished in two stages: First, the separation of the crude or raw material into its constituent parts; second, the purification of these parts so as to render them fit for use and, hence, salable.

71. In a handbook like the present volume, the subject of refining the various crude oils can only be discussed in a broad and general way, as the different crudes require different treatment. No two refineries use exactly the same process, because each oil has its own particular properties and presents its own particular problems for the refiner.

72. The treatments required for the various distillates will also vary with the properties of the crude stock. This must be kept in mind when reading any description of a particular refining process. The **heavy asphalt-base oils** of California and Mexico are usually "**topped**"; that is, only the lighter distillates, such as gasoline, are run off, and the remainder is sold as fuel oil. For this it is actually better than the "whole" crude, as the absence of the more volatile parts makes the handling much safer, so far as risk of fire or explosion is concerned.

73. The **mixed asphaltic paraffin-base oils** of the mid-continent, and the **paraffin-base oils** of the Pennsylvania field, are generally run down to the lubricating oils and the wax.

74. When **refining** paraffin-base crude **by fire and steam** distillation, the oil is charged into horizontal stills heated by fire under the still, and perforated steam coils inside near the bottom of the still. When the temperature of the oil in the still is well above the boiling point of water, steam is admitted through the perforated coils to "sweeten" the product and bring over the benzine at a lower temperature.

75. The vapors from the still pass through the vapor lines to the condenser, where they liquefy and flow to the **"look-box"** in the still house. From the look-box the liquid flows to the "run-down" or temporary storage tanks. The still man controls the distillation process by noting the specific gravity of the liquid flowing through the look-box in the still house. As the specific gravity of the liquid changes, the still man changes the flow of the distillate to its proper run-down tank by means of valves on the look-box manifold.

76. The order in which the various **"cuts"** come over, the vapor temperature and the specific gravity are shown by the following table:

Cut	Vapor Temperature Degrees F.	Baumé Deg.
1st. Crude Benzine	410	46.5–47.
2nd. Kerosene Distillate	572	38.–42.
3rd. Gas Oil	...	36.5
4th. Wax Distillate	...	31.–33.

77. The residue in the still is known as **steam-refined cylinder stock.** This is pumped out through a pipe-cooler to a storage tank.

78. Crude **benzine** includes all the light distillates which vaporize up to 410 deg. F. From this crude benzine we derive both **gasoline** and **naphtha.** In some cases it is only necessary to redistill the crude benzine in steam stills. Other cases require treatment with acid and caustic soda, after which the benzine is thoroughly washed with water. In many cases a good dephlegmator on the crude still makes a marketable gasoline without either the acid treatment or redistilling with steam.

79. Kerosene or water-white distillate is treated with acid and caustic in the agitator and exposed to heat, air and light in a shallow tank or bleacher, in which all the water is settled out. After treatment, if the kerosene is not water-white, or has too high an "end-point," it is redistilled with superheated steam.

80. The **end-point** is the highest temperature reached in distilling off completely a sample of the liquid tested. The U. S. Government specifications define it thus: "The maximum boiling point or dry-point determined by continuing the heating after the flask bottom has boiled dry until the column of mercury (c.c. of the thermometer) reaches a maximum and then starts to recede consistently."

81. The **gas oil** is steam-reduced to remove any kerosene distillate. The gas or fuel-oil residue is sold to the manufacturer of water-gas to "fatten" it, or as a light fuel oil. Gas oil is also used as a scrubber oil in the manufacture of natural-gas gasoline. It is also used as a cracking stock yielding sometimes as high as 65 per cent gasoline.

82. The first operation in **refining the wax** distillate is re-running in fire still with a small amount of open steam. In refining slang, this is called "cracking the amorphous wax." This process "cuts" the lower-boiling products, and the benzine, kerosene or fuel oil are added to their respective cuts from the crude oil. The still residue is added to the wax distillates from the crude. The "cracked" wax distillate is run to refrigerating machines, cooled to 15 deg.-30 deg. F., then pumped to the filter-press and filtered through canvas at a pressure of 300 to 400 lb. per sq. in. The cake formed is known as **"slack wax,"** and the oil squeezed out from the filter-press is known as "pressed distillate."

83. The **slack wax** is melted and then pumped to the sweat-pans. The sweat-pans are of steel, fitted with a galvanized iron or brass screen, and with pipe coils above the screen through which either hot or cold water may be circulated. The pans are about 8 ft. by 20 ft. in area and 14 in. deep. They are filled with cold water to a height of ¼ in. above the screens, after which the melted wax is pumped in. Cold water is then circulated through the coils to solidify the wax. As soon as the wax solidifies, the water is drawn off, and the cake of wax and oil settles on the screen. The temperature inside the sweat-room is now carefully raised, and the oil slowly separates or "sweats" from the wax and drips into the pan. During the sweating operation, two grades of oil are recovered: that from the first part, known as **"footes"** is added to the wax distillate; that from the latter part, known as **"intermediate wax,"** is added to the slack wax. After sweating, the wax remaining is melted by circulating hot water through the pipe coils in the pan, and filtered through fuller's earth or bone char. It is then molded into cakes and is ready for the market.

84. The residue or steam-refined cylinder stock is refined into different grades of **lubricants** as market conditions warrant. The methods employed are too lengthy to be discussed here.

85. The average results obtained by **refining a paraffin-base petroleum** are given in the following table:

	PER CENT
Gasoline and naphtha	30.0
Burning oils	17.5
Fuel and gas oils	22.5
Light lubricating oils	3.5
Medium lubricating oils	4.5
Paraffin wax	1.3
Petrolatum	0.8
Cylinder Oils	15.0
Loss	4.9
	100.0

86. The average results obtained by **refining the mixed-base oils** by both cracking and straight distillation are given in the following table:

	CRACKING DIST. PER CENT	STRAIGHT DIST. PER CENT
Gasoline	25–35	20–30
Burning Oils	15–25	8–12
Gas and Fuel Oils	30–40	40–50
Lubricating Oils	2–5	2–6
Wax Tailings	1	.....
Coke	4–5	.....
Miscellaneous	.....	5–10
Loss	4–6	4–5

87. The results obtained from the **asphaltic-base California oils** are given in the following table:

		DEG. BAUMÉ	PER CENT
Gasoline stock or "top"	Gravity	52–54	13.5
Kerosene " " "slop"	"	37–39	5.0
Stove and furnace distillates	"	30–40	10.0
Residuum or fuel oil	"	17–20	71.5
			100.0

88. The expression or term **"cracking"** referred to above, and met with constantly in petroleum refining, signifies so many different and more or less elaborate processes, that no attempt will be made here to discuss it in detail. But its general significance is simple; it means breaking up an oil chemically by heating it to a very high temperature. As previously stated, crude petroleum is a mixture of compounds. Straight distillation separates the compounds of different boiling points from each other, but does not break them up chemically.

89. If straight distillation yields a relatively small amount of gasoline and burning oils, and a relatively large amount of gas and fuel oils; and if the market price of the gasoline and burning oils is considerably higher than that of the gas and fuel oils, it can easily be seen that any process that would increase the proportion of the higher-priced oils and lessen the proportion of the lower-priced oils would be eagerly sought for, provided such a process did not increase the cost of refining beyond economical limits. This gave rise to the various cracking processes, most or all of which are patented, and known by the names of the several patentees.

90. There are a number of **cracking processes** and various patents. The cracking may be carried on in the vapor phase, at atmospheric

pressure or with increased pressure. Also, there may be cracking in the liquid phase, involving distillation at either atmospheric pressure or above atmospheric pressure, such as the **Burton** process, or at a very high pressure. Also, there may be cracking in the liquid phase without distillation and with high pressure; either without vapor space for equilibrium, as in the continuous processes; or with vapor space, the latter being either intermittent or continuous.

91. In addition to the Burton process, mentioned above, others are the **Hall** process, the **Cross** process, the **Dubbs** process, the **Rittman** process, etc. One of the most recent is that of **Bacon, Brooks and Clark,** patented March, 1920. This process claims a very high recovery of gasoline, giving the following results from heavy crudes with the process operating normally:

OIL TREATED	BAUMÉ GRAVITY	PER CENT 56 DEG. BAUMÉ GASOLINE RECOVERED
Oklahoma gas oil	32	45
Mexican fuel oil	12	50
California fuel oil	14	47
Caddo heavy crude	12–14	48

92. For further interesting information on this topic, the reader is referred to Battle's "Industrial Oil Engineering," Section 4c, pp. 168-186 inclusive.

93. Another rich source of gasoline is the so-called **casing-head gas,** flowing from an oil well between the tubing and the casing. This gasoline may be extracted by either the absorption or the compression method. The absorption method is usually employed where the gasoline recovered is less than one gallon per 1000 cu. ft. of gas. The gas is passed through an oil heavier than gasoline, using this heavier oil as an absorber, and then distilling the absorbed gasoline from the oil by ordinary fractional distillation.

94. Richer casing-head gas is treated by compressing it to 250 lb. per sq. in. pressure, using a two-stage compressor. The first-stage pressure ranges from 25 to 50 lb. per sq. in. and at this pressure, with corresponding cooling, the heavier condensates are "squeezed out" and removed. The non-condensed gas then passes to the

second stage, where it is compressed to 250 lb. per sq. in. at which pressure the lighter and more volatile condensates are recovered. These are then blended with the heavier products of the first stage condensation or with refinery naphtha, until the desired specific gravity is obtained.

REFINERY PUMPS

101. Having now considered the main points in the refining of oil, it is more than ever plain what an important part in refining work is played by the **pumping machinery.** Practically every type of pump manufactured by Worthington—steam-driven, power-driven, displacement and centrifugal—is used somewhere in the refinery. Some parts of the refinery work, like the pumping of light distillates, can be done with the ordinary types of pumps with the proper fittings for the service, while other parts of refinery work, like pumping the heavy crudes, charging the stills, and recirculating hot oils in cracking processes, require special pumps. It is the special pumps and the service to which they are adapted that we will now consider.

102. Block-valve pumps, piston-pattern, are **recommended** for the pumping of hot or cold oils or other thick liquids. This pump derives its name from the shape of the liquid valves, which are in the form of rectangular blocks. The block-valve pump has numerous uses where hot liquids are to be handled at temperatures up to 600 deg. F. It is used for handling hot asphalt from stills through heat exchangers, or coolers, to tanks; for circulating hot oils through coils or stills; for transferring hot residuums from stills to pre-warmers, or to any place where the heat from the residuums may be utilized. Block-valve pumps may also be used for pumping hot tar and other viscous semi-liquids.

103. Block-valve **liquid ends** are of the submerged piston-pattern. The liquid end of the 12-in. stroke pump is of the twin type, with removable discharge valve deck. The suction deck is cast integral with the liquid cylinder. Owing to the size of the liquid cylinder the twin casting is the most economical. The gaskets for the valve plate and force-chamber cover are not unduly large and will not be a source of trouble when the pump is handling liquids at 600 deg. F.

104. The liquid ends of the 18-in. stroke pumps are of the **separate-cylinder type.** This design permits casting the top or force chamber closed, eliminating the large joints necessary in large cylinders of the valve-plate type. With this design all gaskets are of small size, a very desirable feature for pumps handling very hot liquids.

105. The **valve decks** are cast integral with the liquid cylinders. The valve seats are cast integral with the valve decks to prevent trouble arising from seats coming loose from unequal expansion due to temperature changes when the pumps are used for intermittent service on very hot liquids. Liquid valves are accessible through large hand-holes having flanged covers.

106. The liquid-cylinder **liners** are of the flanged removable type. They may be either cast iron or bronze, as the service requires.

107. The rectangular valves work in guides cast on the inside of the hand-hole covers, thus eliminating guards or stems. These guides are machine finished to insure the true seating of the valves. The rectangular valve gives the maximum valve area for space occupied, and offers no obstruction in the form of seat arms or center hub to the free flow of the liquid.

108. When pumping liquids below 400 deg. F., semi-elliptical steel **valve springs** are fitted into slots on top of the valves; the top central portion of the spring bears on the top wall or cover of the liquid cylinder, and its own tension tends to hold it in place. At temperatures above 400 deg. F. steel springs lose their elasticity and when temperatures above this are encountered the valves are made heavy enough to seat without springs.

109. Two types of **stuffing boxes** are used on the liquid end of the block-valve pumps. For cold liquids a stuffing box of the standard studded-gland type is used. For hot liquids an extra deep stuffing box with lantern gland similar to that described in par. 127 to par. 131, inclusive, is used.

110. The **features** of the **block-valve pump** which appeal to the refinery engineer are:

(1.) Minimum number of interior parts; 8 valves, 2 liners and 2 pistons.

(2.) No separate valve seats or guards.

(3.) Unobstructed liquid passages of large area.

(4.) Minimum number of joints and gaskets.
(5.) Minimum weight, maximum strength, low first cost.

111. Worthington Block-valve Pumps with simple duplex steam ends are listed in par. 115. Special combinations of liquid ends and compound steam ends can be made when economy of operation is an important consideration.

112. Hot-oil pumps for cracking processes at temperatures up to 1000 deg. F. and discharge pressures up to 1000 lb. per sq. in. are shown by Fig. 8 and 9. To operate successfully under these abnormal conditions, the liquid end of the pump must be designed and constructed with unusual care in order to withstand, without leaking, the distortion and deflection caused by expansion and contraction due to extreme temperature changes. The materials used in the liquid end must be carefully selected to withstand safely the great reduction in both elasticity and tensile strength that takes place at temperatures of 700 and 1000 deg. F.

113. Liquid ends of either cast iron or cast steel when subjected to high pressures are not dependable for liquids at temperatures above 600 deg. F. The tensile strength of either metal is uncertain; a slight increase in temperatures at these levels being accompanied by a considerable drop in the elastic limit of cast steel. Suitable castings of a uniform quality and thickness are difficult to obtain, especially when cored passages are essential. When pouring castings it is almost impossible to keep cores from shifting. Even a slight shifting of cores results in thin sections or walls.

114. EFFECTS OF TEMPERATURE ON THE MAXIMUM ALLOWABLE PRESSURE ON THE DISCHARGE OF CAST-IRON LIQUID ENDS

100° F.	200° F.	300° F.	400° F.	500° F.	550° F.	600° F.	650° F.	700° F.
Allowable pressures in pounds per square inch								
75	75	75	75	75	70	55	25	0
150	150	150	150	150	135	100	75	0
200	200	200	200	200	180	140	100	0
300	300	300	300	300	270	210	150	0
500	500	500	500	500	450	350	250	0
800	800	800	800	800	720	560	400	0

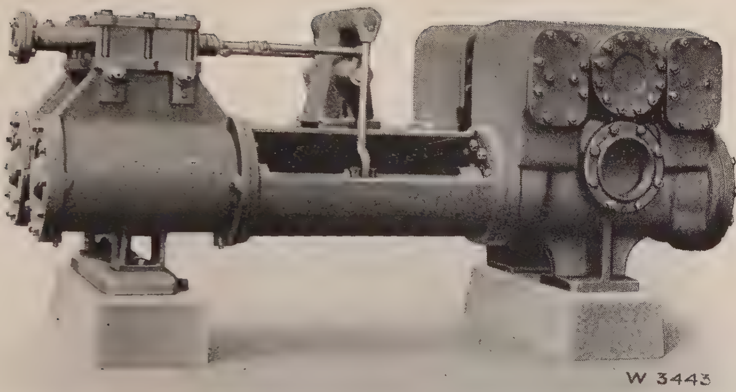


Fig. 6-7.

WORTHINGTON DUPLEX PACKED-PISTON PUMPS
Block-Valve Type

Maximum working pressure: Steam end—150 lb.
Liquid end—See table.

115. TABLE OF SIZES AND DATA

Size of Pump in Inches			**Maximum Discharge Pressure Good for	Capacity for Continuous Operation*					Pipe Sizes, Inches				Floor Space, Inches
Diameter of Steam Cylinders	Diameter of Liquid Cylinders	Stroke		Gal. per Min.	Barrels of 42 Gal. per Hour	Barrels of 42 Gal. per 24 Hours	Piston Speed Ft. per min.	Rev. per min.	Steam	Exhaust	Suction	Discharge	
12	6	12	300	115	164	3,936	40	20	2½	3	5	4	33 x 79
12	7	18	300	233	333	7,992	60	20	2½	3	6	5	48 x 105
14	7	18	300	233	333	7,992	60	20	2½	3	6	5	48 x 105
16	7	18	300	233	333	7,992	60	20	2½	3½	6	5	48 x 106
18	7	18	300	233	333	7,992	60	20	3	4	6	5	48 x 107
12	9	18	300	384	549	13,176	60	20	2½	3	8	6	54 x 108
14	9	18	300	384	549	13,176	60	20	2½	3	8	6	54 x 108
16	9	18	300	384	549	13,176	60	20	2½	3½	8	6	54 x 110
18	9	18	300	384	549	13,176	60	20	3	4	8	6	54 x 112
20	9	18	300	384	549	13,176	60	20	4	5	8	6	54 x 112

* Hot oil or fuel oil. ** Cast steel liquid ends good for 600 lb. discharge pressure can be furnished for special conditions of service.

FIG. 8

WORTHINGTON PISTON-PATTERN SIMPLEX HOT-OIL PUMP
SIZE 16x9x24, WITH FORGED-STEEL LIQUID END

117. Under ordinary conditions of pressure and temperature, the factor of safety is so high that no harm can result from such shifting of cores; but for high pressures and temperatures combined, the slightest shifting renders the casting unfit for use. Steel castings, especially in the larger sizes, are naturally porous to a degree that renders them practically useless for the high temperatures and pressures encountered in hot-oil work.

118. For temperatures above 600 deg. F., however, it has been found that **forged steel** is perfectly dependable as to its behavior under the combined effects of temperature and pressure. Its tensile strength and elastic limit are consistent with temperature changes, the metal is uniform in quality and density, and no trouble is experienced from porosity. Because of this reliability, forged steel has been adopted by Worthington as the material to be used in the liquid ends of hot-oil pumps for high temperatures combined with high pressures.

119. The **steam ends** may be of either the simplex or duplex type having steam valves of either the slide or piston pattern. **Duplex** steam ends are always fitted with cushion valves, as it is essential that the stroke be closely regulated.

120. The liquid end is an entirely new type and has been designed to meet the most severe service conditions. The liquid end is of the submerged piston type, the **simplex** being made from one solid forged-steel billet. The **duplex** liquid end is made from one solid forged-steel billet whenever the centers permit. This construction eliminates cross suction and cross discharge pipes with their four joints.

121. Service conditions are sometimes encountered where it becomes necessary to use large-diameter steam cylinders for the duplex pump. These large steam cylinders are necessarily spaced on centers of such width as to make it impractical to use one forging for the liquid end. When this condition arises, two forgings are used, one for the right-hand cylinder and one for the left-hand cylinder. Suction and discharge Y pipes, designed to take care of the expansion and contraction due to temperature changes, are provided to properly connect the two cylinders to a common suction and discharge pipe.

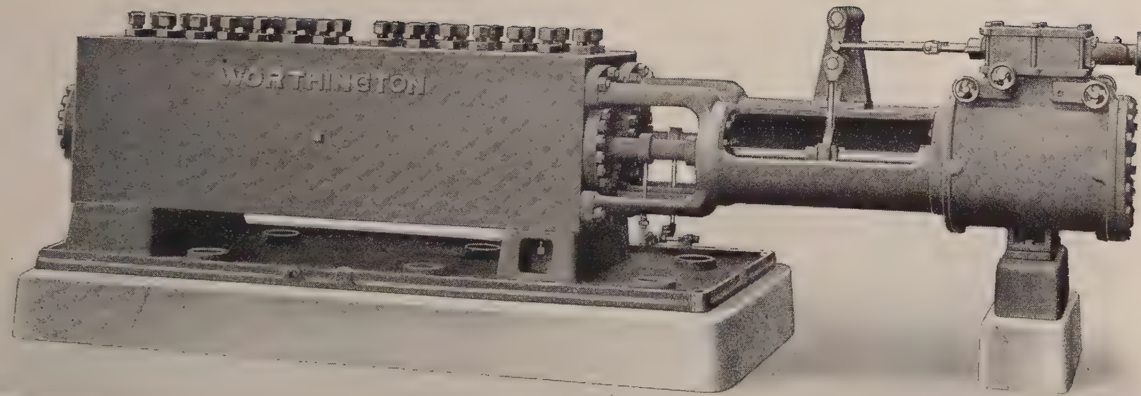


FIG. 9

WORTHINGTON PISTON-PATTERN DUPLEX HOT-OIL PUMPS
SIZE 18 X $8\frac{1}{2}$ X 18, WITH ONE-PIECE FORGED STEEL LIQUID END

122. In both the simplex and duplex types the passages for the liquid pistons are bored from end to end of the cylinder. The cradle end of the passage is then closed by the piston-rod stuffing box and the outboard end by a forged-steel cylinder head. Both the **stuffing box** and the **cylinder head** are bolted to the cylinder with a male and female fitting and the joint made tight by means of aluminum-cased asbestos gaskets.

123. The **ports** for the liquid valves are drilled on each side of the suction and discharge ports, intersecting these two ports as well as the passage for the liquid pistons. All intersections are carefully milled out to prevent the formation of gas pockets in the liquid cylinders.

124. The liquid-cylinder **liners** are of special steel, forced into position and secured by outside adjustable set screws. These set screws have aluminum-cased asbestos gaskets under the heads to prevent leakage.

125. The **liquid valves** are of steel of the **ball type,** working upon steel seats fitted into the cylinder on a taper. The ports for the valves are closed by forged-steel plugs.

126. The **liquid pistons** are of the solid type fitted with special hammered iron snap rings. The pistons are fitted on the rods on a taper and secured by means of forged-steel nuts and lock nuts.

127. For all pumps handling hot oils it is absolutely necessary to make the piston and **stuffing boxes** on the liquid end of special design to permit the hot oil from escaping into the air and taking fire. Special stuffing boxes are also used when the oil is pumped at a temperature which may be below its flash point but at the same time may be high enough to cause objectionable smoking should the hot rod come into the air.

128. The **Worthington Hot-Oil Stuffing Box** is made extra deep and arranged for a positive cold-oil circulation. The depth of the box is such that no portion of the hot rod can extend into the air. The cold circulating oil is brought into direct contact with the piston rod by means of a lantern gland inside the stuffing box. The circulating oil, together with any hot oil that may leak past the packing, is trapped by a second lantern gland located near the

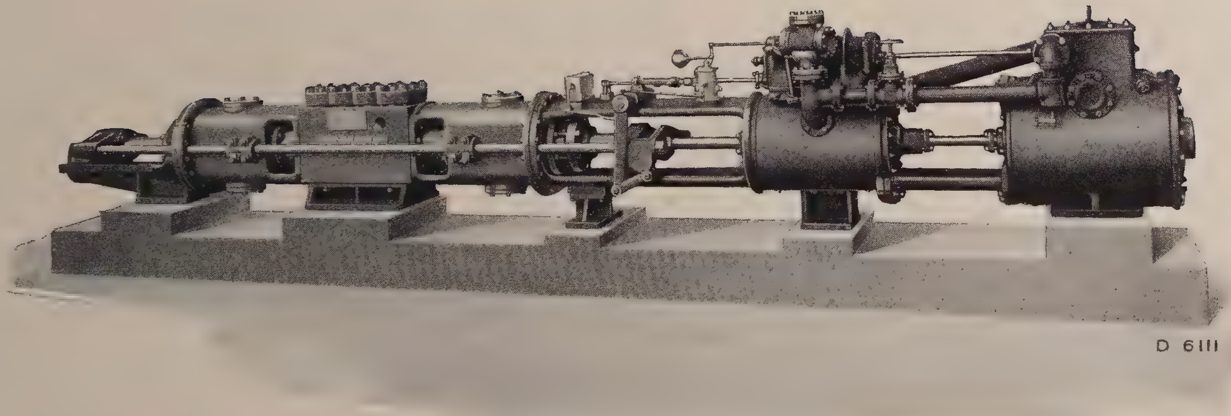


Fig. 10

Worthington Compound Simplex Hot-oil Pumps
Size 16 and 24 by 5 by 24, End-Packed Plunger Pattern
For 1500 lb. Discharge Pressure

gland end of the stuffing box, from which point any leakage may be piped back to the suction. Cold charging stock, of the same characteristics as the oil being pumped, may also be used as the cooling medium, and the stuffing box arranged so that oil will be drawn into the cylinder on the suction stroke of the pump.

129. When outside-packed plunger pumps are used on hot-oil service, water-jacketed plunger stuffing boxes are employed.

130. Worthington Hot-Oil Pumps, whether for high or low-pressure work, are always equipped with special stuffing boxes on the liquid ends which insure safety of operation under all conditions of service.

131. For all temperatures up to 600 deg. F. two rings of braided copper in the bottom of the box, followed by copper-wire-inserted, braided asbestos rings next to the lantern gland and between the lantern gland and outside gland, have been found to be a very satisfactory method of packing these boxes. For temperatures above 600 deg F., special metallic packing rings are used between the lantern gland and the bottom of the box, and copper-wire, inserted, long-fibre-braided asbestos between the lantern gland and the outside gland.

132. The liquid-end **piston rods, studs and bolts** are of a special grade of steel, selected for its resistance to distortion when subjected to high temperatures.

133. Operating conditions vary to such an extent that no attempt can be made to list the complete line of hot-oil pumps.

134. A 16 by 9 by 24 **simplex piston-pattern pump,** rated capacity 10,000 bbl. per 24 hr. is shown by Fig. 8, and an 18 by 8½ by 18 duplex piston-pattern of 12,000 bbl. per 24 hr. capacity, by Fig. 9. Both of these pumps are good for all temperatures up to 1000 deg. F. and all discharge pressures up to 1000 lb. per sq. in.

135. A **compound simplex double-plunger pump** is shown by Fig. 10. This pump is rated at 3250 bbl. per 24 hr. of 1000 deg. F. oil against 1200 lb. per sq. in. discharge pressure.

136. Worthington Hot-Oil Pumps, piston-pattern, are standardized as to liquid end sizes for **capacities** of 3500, 5000, and 10,000 bbl. per 24 hr. Steam ends are available in all sizes, both simple and compound, for all conditions of service.

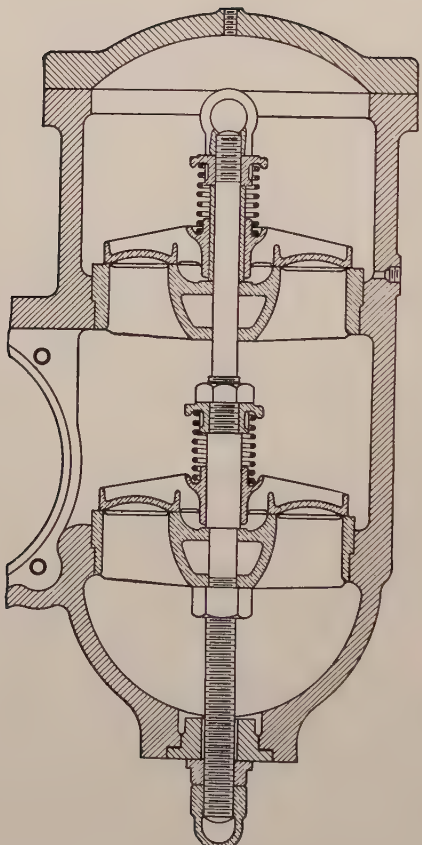


Fig. 11

Worthington Cargo Oil Pump.
Liquid Valve Service.

137. When oil is needed in quantities at places not served by pipe lines, it is transported by tank cars, if reached most easily by land; or by tankships or tank barges if reached more easily by water. Tank cars and tank ships are usually loaded at places specially adapted for this purpose, known as loading stations. When pumping is necessary for the quick loading and unloading of tankers or barges, the Worthington Horizontal Duplex **"Cargo Oil" Pump** has been found to be very satisfactory for this service.

138. As "time idle" is an important item in the cost of operating tankers, it is necessary for loading and unloading to use large powerful pumps that are simple and dependable, in order to make a quick "turnabout." These pumps must have exceptionally large valve area, all working parts must be easy of access, and must require a minimum of attention. The Worthington "Cargo Oil" Pump, table, par. 141, has been designed to meet these requirements.

139. The **liquid valve service,** Fig. 11, illustrates the care and attention given to accessibility of the working parts of these pumps. The entire suction and discharge valve assembly of one section can be removed from the pump and a new assembly inserted quickly and easily. The valve-chamber cover and the nuts are removed and the entire valve assembly, including the suction and discharge valves, valve seats, stem, guards, and springs lifted out by means of the eye bolt. All other working parts are designed for the same ease of removal or replacement so that repairs or adjustments can be made quickly in case of emergency either at sea or in harbor.

140. The Worthington Duplex Submerged Piston Pump, **"Refinery" pattern,** listed in table, par. 142, is extensively used on tank barges for unloading fuel oils, kerosene, gasoline, etc. This pump can be equipped with either disk or ball valves, separate removable valve decks or cast-in valve decks. This pump also has many uses in the refinery for pumping water, light oils or distillate.

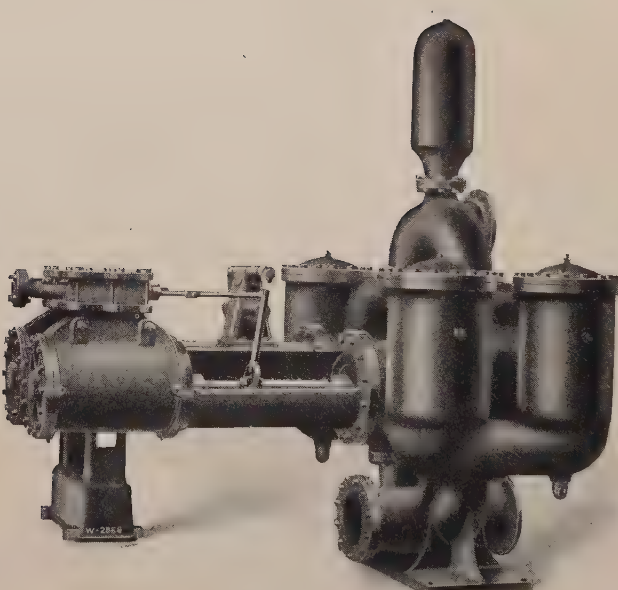


FIG. 12.

WORTHINGTON DUPLEX PISTON PUMP

Cargo Pattern

WORTHINGTON DUPLEX PISTON PUMPS

Cargo Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end—See table.

Fig. 12.

When conditions require, these pumps can be furnished with compound steam ends of suitable sizes to meet the service conditions.

141. TABLE OF SIZES AND DATA

Size of Pump in Inches: Diameter Steam Cylinders	Size of Pump in Inches: Diameter Liquid Cylinders	Size of Pump in Inches: Stroke	Maximum Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Capacity: Barrels of 42 Gal. per Hour	Capacity: Barrels of 42 Gal. per 24 Hours	Capacity: Piston Speed Ft. per min.	Capacity: Rev. per min.	Pipe Sizes, Inches: Steam	Pipe Sizes: Exhaust	Pipe Sizes: Suction	Pipe Sizes: Discharge	Floor Space Ft. and In.
12	10	18	250	374	534	12,816	48	16	2½	3	10	8	9-0 x 5-8
14	10	18	250	374	534	12,816	48	16	2½	3	10	8	9-0 x 5-8
16	10	18	250	374	534	12,816	48	16	2½	3½	10	8	9-2 x 5-8
12	10½	18	250	413	590	14,160	48	16	2½	3	10	8	9-0 x 5-8
14	10½	18	250	413	590	14,160	48	16	2½	3	10	8	9-0 x 5-8
16	10½	18	250	413	590	14,160	48	16	2½	3½	10	8	9-2 x 5-8
12	11	18	250	450	643	15,432	48	16	2½	3	10	8	9-0 x 5-8
14	11	18	250	450	643	15,432	48	16	2½	3	10	8	9-0 x 5-8
16	11	18	250	450	643	15,432	48	16	2½	3½	10	8	9-2 x 5-8
16	12	18	250	535	764	18,336	48	16	2½	3½	12	10	10-5 x 6-11
16	13	18	250	630	900	21,600	48	16	2½	3½	12	10	10-5 x 6-11
16	14	18	250	735	1050	25,200	48	16	2½	3½	12	10	10-5 x 6-11
16	12	24	250	625	893	21,432	56	14	2½	3½	12	10	11-6 x 6-11
16	13	24	250	740	1057	25,368	56	14	2½	3½	14	12	11-6 x 6-11
18	13	24	250	740	1057	25,368	56	14	3	4	14	12	11-7 x 6-11
16	14	24	250	855	1221	29,304	56	14	2½	3½	14	12	11-6 x 6-11
18	14	24	250	855	1221	29,304	56	14	3	4	14	12	11-7 x 6-11
20	14	24	250	855	1221	29,304	56	14	4	5	14	12	11-7 x 6-11
25	16½	24	300	1180	1686	40,464	56	14	5	6	18	14	12-3 x 10-10

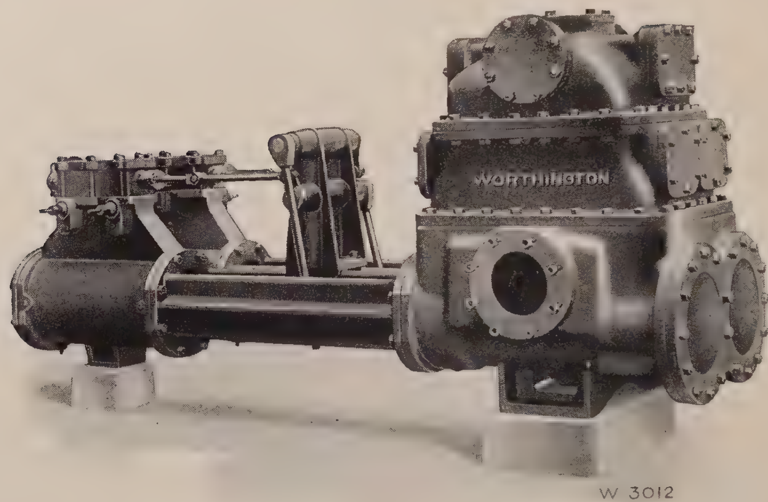


Fig. 13.

WORTHINGTON DUPLEX PACKED-PISTON PUMPS

"Refinery" Pattern

Maximum working pressure: Steam end—150 lb.
Liquid end—150 lb.

142. TABLE OF SIZES AND DATA

Size of Pump in Inches			Capacity for Continuous Operation						Pipe Size, Inches				Floor Space, Inches
			Water or Light Oils			Heavy Oils							
				Barrels of 42 Gal. Per			Barrels of 42 Gal. Per						
Diameter of Steam Cylinders	Diameter of Liquid Cylinders	Stroke	Gal. per min.	Hour	24 Hours	Gal. per min.	Hour	24 Hours	Steam	Exhaust	Suction	Discharge	
12	8	18	455	650	15,600	245	350	8,400	2½	3	8	6	109 x 35
12	9	18	575	822	19,728	310	443	10,632	2½	3	8	6	109 x 35
12	10	18	710	1,015	24,360	380	543	13,032	2½	3	8	6	109 x 35
14	8	18	455	650	15,600	245	350	8,400	2½	3	8	6	109 x 40
14	9	18	575	822	19,728	310	443	10,632	2½	3	8	6	109 x 40
14	10	18	710	1,015	24,360	380	543	13,032	2½	3	8	6	109 x 40

The capacity for water or light oils is calculated for a piston speed of 60 ft. per min. or 20 r.p.m. For heavy oils the capacity is calculated for a piston speed of 48 ft. per min. or 16 r.p.m.

150. Oil-line pumps, whether for gathering lines, feeder lines or trunk lines, must be of heavy, rugged construction, capable of continuous operation at pressure of 500 to 1000 lb. per sq. in. For economy in operation the oil passages must be short, straight and of large area, in order to keep friction losses through the pump down to the minimum.

151. The duplex piston pumps listed in table, par. 155 are used extensively for gathering pumps and for general field work. They can be mounted on skids, and being of medium weight can be easily moved from lease to lease as required. The smaller sizes of **"California" pattern** pumps, table, par. 156, are also used for gathering lines as well as for small feeder lines. The larger sizes are suitable for small trunk-line pumping.

152. Large trunk lines require permanent, well designed pumping stations with either steam-driven or power-driven pumps ranging in capacity from 10,000 to 30,000 bbl. per day. These pumps are always built to meet some special requirements, and no attempt will be made to list the sizes and capacities. Fig. 14 illustrates a steam-driven pump for 30,000 bbl. per day. Steam-driven pumps are built with compound or triple-expansion steam ends. When pumping crudes having high viscosities at atmospheric temperatures, the exhaust steam from the pumping engine is used to heat the oil and reduce the viscosity to a point where the oil will be sufficiently fluid to be pumped through the pipe line.

153. Power pumps similar to Fig. 15 **driven by Worthington Diesel Engines** have been used for some of the more recent trunk-line installations. These are very economical units. With fuel oil at $3.00 per bbl. and lubricating oil at 50 cents per gal., a Diesel-engine-driven pump can be operated against a line pressure of 700 lb. per sq. in. at a cost of $1.55 per day per 1000 bbl. pumped.

154. Power-driven oil-line pumps are available for all capacities up to 35,000 bbl. per day.

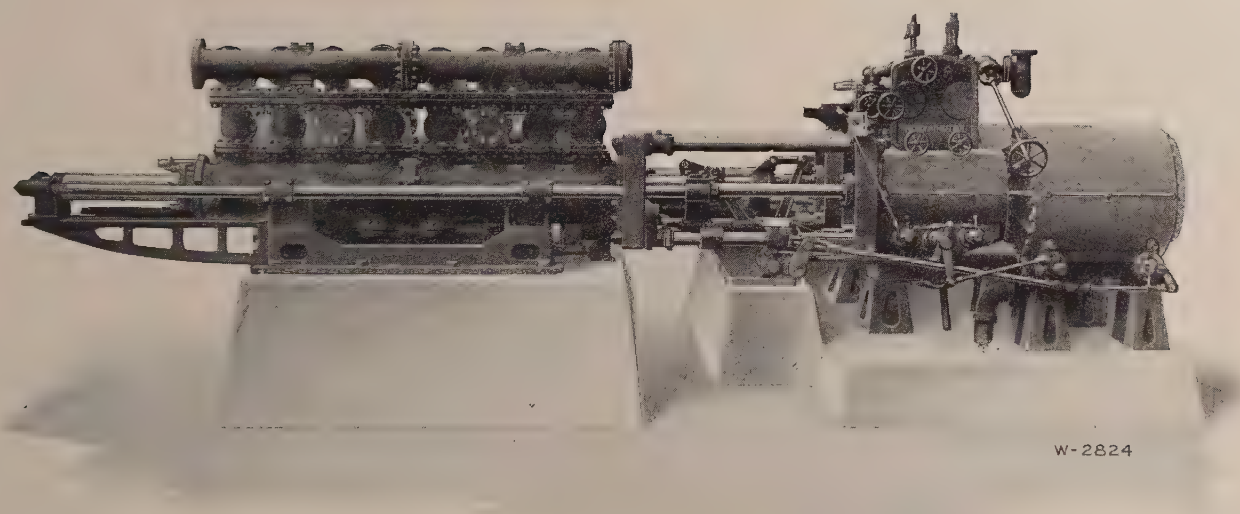


FIG. 14

WORTHINGTON DUPLEX COMPOUND OIL-LINE PUMP. END-PACKED PLUNGER PATTERN.
SIZE 25 AND 42 BY $9\frac{1}{2}$ BY 36.

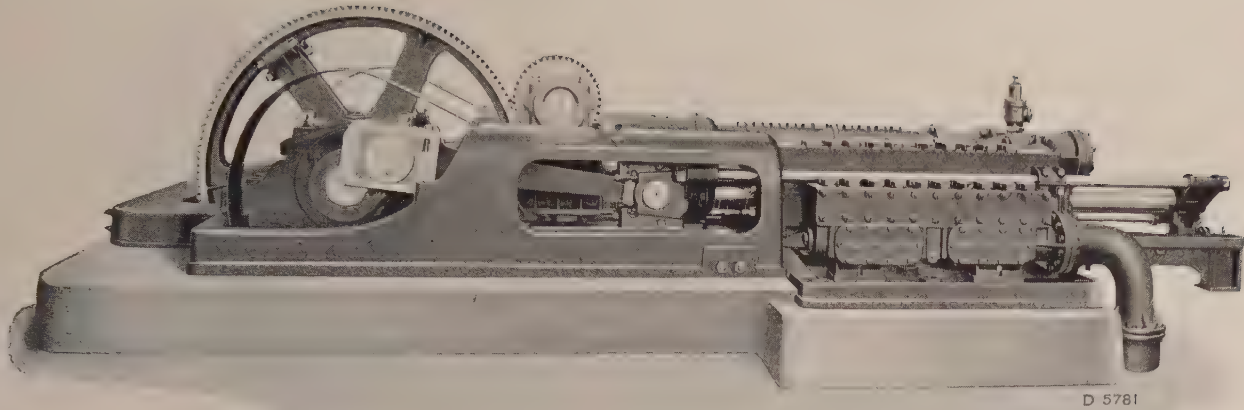


Fig. 15

Worthington Duplex Oil-Line Pump. End-Packed Plunger Pattern Size $6\frac{1}{2}$ by 24 For Direct-connection to Worthington Diesel Engine.

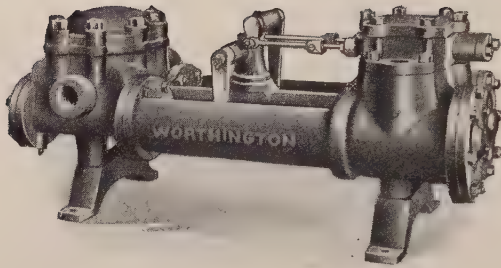


Fig. 16.

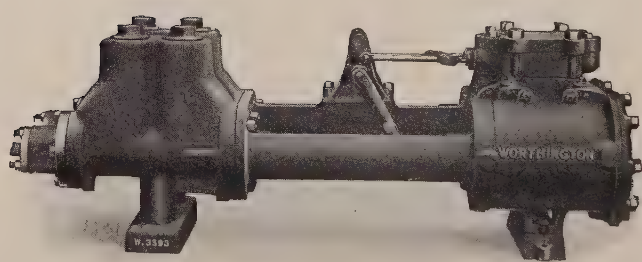


Fig. 17.

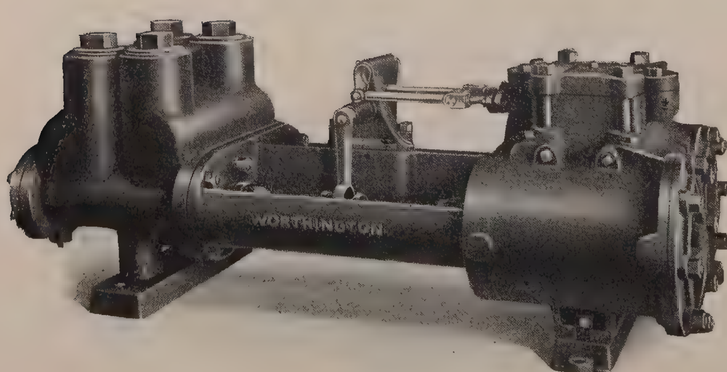


Fig. 18.

Worthington Duplex Packed-Piston Pumps

WORTHINGTON DUPLEX PACKED-PISTON PUMPS

Maximum working pressure: Steam end —150 lb.
Liquid end—See table.

155. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump: Diameter Steam Cylinders	Size of Pump: Diameter Liquid Cylinders	Size of Pump: Stroke	Maximum Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Barrels of 42 Gals. per Hour	Barrels of 42 Gals. per 24 Hours	Piston Speed Ft. per Min.	Rev. per min.	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space, Inches
16	6	2½	6	500	13	19	456	26	26	1	1½	2½	2	45 x 17
16	6	3	6	500	18	26	624	26	26	1	1½	2½	2	45 x 17
16	6	3½	6	500	25	36	864	26	26	1	1½	2½	2	45 x 17
16	7	2½	8	500	15	21	504	30	23	1¼	2	2½	2	51 x 19
16	7	3	8	500	21	30	720	30	23	1¼	2	2½	2	51 x 19
16	7	3½	8	500	29	42	1008	30	23	1¼	2	2½	2	51 x 19
17	10	3	10	600	27	39	936	38	23	1½	2	3	2	70 x 22
17	10	3½	10	600	37	53	1272	38	23	1½	2	3	2	70 x 22
18	10	4½	10	600	62	89	2136	38	23	1½	2	4	2	71 x 22
18	10	4	12	1000	50	72	1728	39	20	2	2½	5	4	79 x 31
18	12	4	12	1000	50	72	1728	39	20	2	2½	5	4	81 x 31
18	14	4	12	1000	50	72	1728	39	20	2½	3	5	4	83 x 31
18	10	4½	12	1000	63	90	2160	39	20	2	2½	5	4	79 x 31
18	12	4½	12	1000	63	90	2160	39	20	2	2½	5	4	81 x 31
18	14	4½	12	1000	63	90	2160	39	20	2½	3	5	4	83 x 31
18	10	5	12	1000	78	112	2688	39	20	2	2½	5	4	79 x 31
18	12	5	12	1000	78	112	2688	39	20	2	2½	5	4	81 x 31
18	14	5	12	1000	78	112	2688	39	20	2½	3	5	4	83 x 31

The capacities given above are for oils of 10° to 20° Baumé with a minimum temperature of 60° F. For heavier oils, speeds must be reduced in accordance with the character of the oil handled.

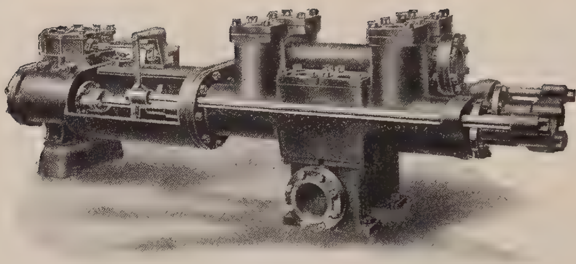


Fig. 19.

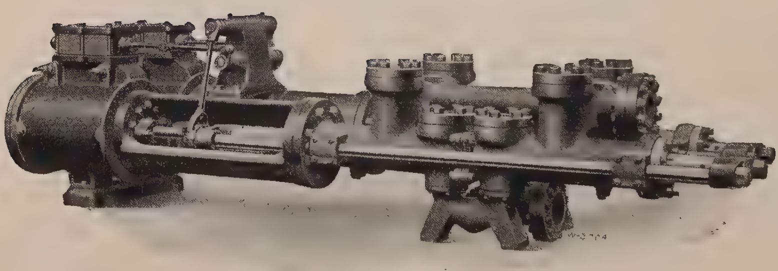


Fig. 20.

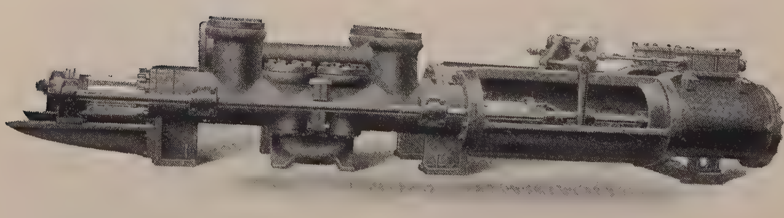


Fig. 21.

Worthington Duplex End-Packed Plunger Pumps

California Pattern

WORTHINGTON DUPLEX END-PACKED PLUNGER PUMPS

"California" Pattern

Maximum working pressure: Steam end —150 lb. per sq. in.
Liquid end—See table.

When conditions require, these pumps can be furnished with compound steam ends of suitable sizes to meet the service conditions.

155. TABLE OF SIZES AND DATA

Fig. No.	Size of Pump, Inches: Diameter Steam Cylinders	Diameter Liquid Cylinders	Sotker	Maximum Discharge Pressure Good for	Capacity for Continuous Operation: Gal. per min.	Barrels of 42 Gal. per One Hour	24 Hours	Piston Speed Ft. per min.	Rev. per min.	Pipe Sizes, Inches: Steam	Exhaust	Suction	Discharge	Floor Space, Inches
19	10	4	12	600	41	59	1416	32	16	2	2½	5	4	128 x 30
19	12	4	12	600	41	59	1416	32	16	2	2½	5	4	132 x 33
19	14	4	12	600	41	59	1416	32	16	2½	3	5	4	132 x 33
19	10	5	12	600	63	90	2160	32	16	2	2½	6	5	131 x 32
19	12	5	12	600	63	90	2160	32	16	2	2½	6	5	131 x 32
19	14	5	12	600	63	90	2160	32	16	2½	3	6	5	131 x 32
19	16	5	12	600	63	90	2160	32	16	2½	3	6	5	134 x 40
19	14	4	18	600	61	87	2088	48	16	2½	3	5	4	167 x 37
19	14	5	18	600	95	136	3264	48	16	2½	3	6	5	169 x 37
19	16	5	18	600	95	136	3264	48	16	2½	3½	6	5	170 x 46
20	18	5	18	1500	95	136	3264	48	16	3	4	6	5	178 x 48
20	20	5	18	1500	95	136	3264	48	16	4	5	6	5	180 x 50
21	16	6	18	600	138	198	4752	48	16	2½	3½	8	6	176 x 46
21	18	6	18	600	138	198	4752	48	16	3	4	8	6	178 x 48
21	20	6	18	600	138	198	4757	48	16	4	5	8	6	179 x 50
21	16	6½	18	600	160	228	5472	48	16	2½	3½	8	6	176 x 46
21	18	6½	18	600	160	228	5472	48	16	3	4	8	6	178 x 48
21	20	6½	18	600	160	228	5472	48	16	4	5	8	6	178 x 50
21	20	7	24	800	186	267	6408	56	14	4	5	8	7	234 x 63
21	22	7	24	800	186	267	6408	56	14	5	6	8	7	246 x 63
21	25	7	24	800	186	267	6408	56	14	5	6	8	7	249 x 67

160. FLOW OF OILS THROUGH COMMERCIAL PIPES

Size of Pipe and Average Inside Diameter, Inches	Capacity, Gallons per Minute	Pressure In Pounds Per Square Inch Per 100 Feet of Pipe Based on Oils of 20° Baumé Gravity — Viscosity in Saybolt Seconds					
		100	200	300	400	500	600
½-in. 0.622	2	6.59	14.3	21.7	29.1	36.9	43.6
	5	21.1	34.8	53.4	71.5	89.5	107.0
	7	37.7	49.2	74.6	101.0	129.0	152.0
	10	69.8	85.8	107.0	144.0	177.0	217.0
	15	143.0	174.0	194.0	217.0	267.0	325.0
¾-in. 0.824	2	2.16	4.74	7.04	9.48	11.6	14.2
	5	5.6	11.5	17.1	23.1	28.9	35.2
	7	9.96	16.1	24.5	32.8	40.5	48.9
	10	18.4	22.3	34.9	46.0	58.5	70.9
	15	37.9	45.7	51.5	70.4	88.3	106.0
1-in. 1.05	5	2.06	4.32	6.68	8.85	11.0	13.4
	10	5.9	8.74	13.4	17.9	22.0	26.6
	15	11.9	14.5	19.7	26.6	33.2	40.3
	20	19.7	23.8	26.8	35.4	44.1	53.5
	25	29.0	35.1	39.3	44.2	54.5	66.8
1½-in. 1.61	10	0.783	1.57	2.40	3.22	3.98	4.86
	20	2.61	3.17	4.85	6.52	8.02	9.70
	30	5.30	6.43	7.15	9.68	12.1	14.7
	40	8.72	10.6	11.9	12.8	16.0	19.4
	50	12.7	15.7	17.4	18.8	18.8	24.2
2-in. 2.067	10	0.266	0.578	0.875	1.17	1.47	1.79
	20	0.79	1.15	1.75	2.34	2.92	3.52
	30	1.60	1.93	2.62	3.50	4.40	5.30
	40	2.63	3.18	3.56	4.66	5.85	7.10
	50	3.86	4.68	5.28	5.88	7.26	8.85
4-in. 4.026	50	0.174	0.198	0.307	0.412	0.494	0.612
	100	0.550	0.668	0.744	0.810	1.01	1.22
	150	1.09	1.36	1.50	1.62	1.74	1.84
	200	1.81	2.22	2.49	2.68	2.85	2.98
	250	2.67	3.26	3.67	3.97	4.20	4.41
6-in. 6.065	100	0.0788	0.0956	0.115	0.158	0.194	0.236
	200	0.259	0.315	0.359	0.388	0.408	0.480
	500	1.27	1.55	1.74	1.88	2.01	2.20
	700	2.29	2.80	3.15	3.36	3.60	3.78
	1000	4.24	5.21	5.82	6.32	6.68	7.05

Concluded on following page.

160. FLOW OF OILS THROUGH COMMERCIAL PIPES—Concluded

Size of Pipe and Average Inside Diameter, Inches	Capacity, Gallons per Minute	Pressure in Pounds Per Square Inch Per 100 Feet of Pipe Based on Oils of 20° Baumé Gravity — Viscosity in Saybolt Seconds 100	200	300	400	500	600
	200	0.0696	0.0834	0.0947	0.102	0.128	0.155
8-in.	500	0.340	0.418	0.459	0.500	0.530	0.556
8.03	1000	1.13	1.37	1.55	1.67	1.78	1.87
	1500	2.27	2.74	3.07	3.30	3.59	3.75
	2000	3.88	4.55	5.15	5.56	5.92	6.22
	1000	0.165	0.203	0.288	0.242	0.258	0.272
12-in.	2000	0.552	0.670	0.750	0.810	0.866	0.906
12.05	3000	1.15	1.34	1.50	1.63	1.73	1.83
	4000	1.01	2.22	2.50	2.69	2.84	3.00
	5000	2.90	3.34	3.68	3.97	4.22	4.41

161. PIPE-LINE PUMPS—SIZES AND TYPES FOR VARIOUS CAPACITIES

Bbl. Per 24 Hr.	Bbl. Per Hr.	Size, Hor. Duplex Double Plunger inches	Speed, r.p.m.	Horsepower at 700 Lb. Per Sq. In.	Suction Size inches	Discharge Size inches	Size, Vert. Triplex Single-Acting inches	Speed, r.p.m.
1000	41.7	....	..	17	2	2	2¾ x 8	50
2000	83.3	....	..	32	3	3	3½ x 10	49
3000	125.	3¼ x 12	53	45	4	3	4¼ x 10	50
4000	167.	3¾ x 12	53	59	4	3	4½ x 12	49
5000	208.	4¼ x 12	52	72	6	4	5 x 12	50
6000	250.	4¾ x 12	50	86	6	4	....	..
8000	333.	4⅝ x 18	47	115	6	4	Horizontal	..
10 000	417.	5½ x 18	41	140	8	6	Triplex	..
12,000	500.	5¾ x 18	45	170	8	6	Dbl. Plunger	..
14,000	583.	5½ x 24	43	195	8	6	....	..
16,000	667.	6 x 24	42	225	8	6	5½ x 18	44
18,000	750.	6¼ x 24	43	250	10	8	5¾ x 18	45
20,000	833.	6½ x 24	44	280	10	8	5½ x 24	41
25,000	1040.	7¼ x 24	45	350	10	8	6 x 24	43
30,000	1250.	6½ x 36	44	420	12	8	6½ x 24	44
35,000	1460.	....	..	490	12	10	7 x 24	45
40,000	1670.	....	..	560	12	10	6½ x 24	45
45,000	1875.	....	..	630	12	10	6½ x 36	44
50,000	2080.	....	..	700	12	10	7 x 36	43

Above pump sizes and speeds suitable for medium and light oils similar to the mid-continent oils. Suitable sizes at slower speeds can be furnished for heavy oils.

$$\text{Horsepower} = \frac{\text{Barrels per hour} \times \text{Pressure}}{2450 \times \text{Efficiency}}$$

162. PIPE LINE DATA

Size Pipe	External Dia. Inches	Internal Dia. Inches	D^2 Sq. In.	Thick-ness Inches	Wt. Per Ft. in Lb.	* Working Press., lb. per sq. in.	† Test Press., lb. per sq. in., Lap Weld.	Bbl. (42 Gal.) per Mile of Pipe	‡ Comparative Discharge. Turb. Flow	‡ Comp. Dis-charge, Straight Line Flow
2	2.375	2.067	4.27	0.154	3.65	1300	1800	21.0	2.7	0.45
3	3.500	3.068	0.41	0.216	7.57	1200	1800	48.2	7.8	2.2
4	4.500	4.026	16.21	0.237	10.79	1000	1600	83.1	16.	6.5
6	6.625	6.065	36.78	0.280	18.97	800	1500	188.5	48.	33.3
8	8.625	7.981	63.7	0.322	28.55	700	1200	326.5	100.	100.
10	10.750	10.020	100.4	0.365	40.48	630	1000	515.0	183.	248.
12	12.750	12.000	144.	0.375	49.56	545	900	738.0	297.	511.

(*) Based on 700 lb. per sq. in for 8-in. pipe which gives a stress of 8700 lb. per sq. in.
(†) National Tube Company.
(‡) Based on equal lengths, pressures and viscosities.

SECTION VIII

Tabular Section

(Figures refer to paragraph numbers)

Tables of Weights and Measures: U. S., 1–9; metric, 10–14; comparative tables, 15; Conversion tables, 16–24; Miscellaneous conversion factors, 25; Miscellaneous equivalents, 26.

American Pipe Standards, cast-iron pipe and flanges, for 125 lb. per sq. in. pressure, 30; for 250 lb. per sq. in., 31.

Fire Protection: Effectiveness of streams, 40–43; Fire-stream tables, 44–54; Nozzle factors, 55; Nozzle or engine pressures, formula, 56; Effective reach of fire streams, 57; Friction loss in fire hose, 58; Discharge table for smooth nozzles, 59; Engine or hydrant pressures for different size nozzles, different size hose and various hose lengths, 60–73.

SECTION VIII
Tabular Section

TABLES OF WEIGHTS AND MEASURES

1. LINEAR MEASURE

				In.	Ft.	Yd.	Rd.	Fur.
12 inches (in.)	= 1 foot	ft.	=	12	1	..	..	.
3 feet	= 1 yard	yd.	=	36	3	1	..	.
5.5 yards	= 1 rod	rd.	=	198	16.5	5.5	1	.
40 rods	= 1 furlong	fur.	=	7,920	660	220	40	1
8 furlongs	= 1 mile	mi.	=	63,360	5,280	1,760	320	8

1 sea mile (U. S.) = 6,080 ft. = 1 1/6 statute miles, roughly

2. SURVEYOR'S MEASURE

7.92	inches	= 1 link	li.
25	links	= 1 rod	rd.
4	rods	= 1 chain	ch.
100	links	= 1 chain	ch.
66	feet	= 1 chain	ch.
80	chains	= 1 mile	mi.

1 Mile = { 80 ch. = 320 rd. = 8,000 li. = 63,360 in.

3. SQUARE MEASURE

144	square inches (sq. in.)	= 1	square foot	sq. ft.
9	" feet	= 1	" yard	sq. yd.
30¼	" yards	= 1	" rod	sq. rd.
160	" rods	= 1	acre	A.
640	" acres	= 1	square mile	sq. mi.

1 sq. mi. = { 640 A. = 102,400 sq. rd. = 3,097,600 sq. yd. = 27,878,400 sq. ft. = 4,014,489,600 sq. in.

4. GUNTER'S CHAIN

Used in land surveying

7.92	inches	=	1 link
100	links	=	1 chain
80	chains	=	1 mile
1	chain	=	4 rods
4	rods	=	66 feet

5. SURVEYOR'S SQUARE MEASURE

625 square links (sq. li.)	=	1 square rod	sq. rd.
16 " rods	=	1 " chain	sq. ch.
10 " chains	=	1 acre	A.
640 acres	=	1 square mile	sq. mi.
36 square miles	=	1 township	tp.

$$1 \text{ sq. mi.} = \begin{cases} 640 \text{ A.} = \\ 6{,}400 \text{ sq. ch.} = \\ 102{,}400 \text{ sq. rd.} = \\ 64{,}000{,}000 \text{ sq. li.} \end{cases}$$

The acre = 4,840 sq. yd. or 43,560 sq. ft., and is equal to the surface of a square measuring 208.7 ft. on a side.

6. CUBIC MEASURE

1728 cubic inches (cu. in.)	=	1 cubic foot	cu. ft.
27 " feet	=	1 " yard	cu. yd.
128 " feet	=	1 cord	cd.
24¾ " feet	=	1 perch	P.
1 cubic yd.	=	27 cu. ft. = 46,656 cu. in.	

7. AVOIRDUPOIS WEIGHT

437.5 grains (gr.)	=	1 ounce	oz.
16 ounces	=	1 pound	lb.
100 pounds	=	1 hundredweight	cwt.
20 cwt. / 2000 pounds	=	1 ton	T.

$$1 \text{ T.} = \begin{cases} 20 \text{ cwt.} = \\ 2{,}000 \text{ lbs.} = \\ 32{,}000 \text{ oz.} = \\ 14{,}000{,}000 \text{ gr.} \end{cases}$$

1 lb. avoirdupois = 7,000 gr.

8. LONG TON TABLE

16 ounces	= 1 pound	lb.
112 pounds	= 1 hundredweight	cwt
20 hundredweight 2240 pounds	= 1 ton	T.

9. LIQUID MEASURE

4 gills (gi.)	= 1 pint	pt.
2 pints	= 1 quart	qt.
4 quarts	= 1 gallon	gal.
31½ gallons	= 1 barrel	bbl.
2 barrels 63 gallons	= 1 hogshead	hhd.

1 hhd. = { 2 bbl. = 63 gal. = 252 qt. = 504 pt. = 2,016 gi.

The U. S. gallon contains 231 cu. in. = 0.134 cu. ft. or 1 cu. ft. contains 7.481 gal.

When water is at its maximum density 1 cu. ft. = 62.425 lb. and 1 gal. weighs 8.345 lb.

The British imperial gallon, both liquid and dry, contains 277.274 cu. in. = 0.16046 cu. ft. and is equivalent to the volume of 10 lb. of pure water at 62 deg. F. To reduce British imperial to U. S. liquid gallons, divide by 1.2. To convert U. S. liquid gallons to British imperial multiply 1.2 or increase the number of U. S. gallons by 1/5.

10. METRIC SYSTEM

Measures of length

10 millimeters (mm.)	= 1 centimeter	cm.
10 centimeters	= 1 decimeter	dm.
10 decimeters	= 1 meter	m.
10 meters	= 1 decameter	dm.
10 decameters	= 1 hectometer	hm.
10 hectometers	= 1 kilometer	km.
10 kilometers	= 1 myriameter	Mm.

The meter is defined as one ten millionth of the distance from the pole to the equator, measured on a meridian passing near Paris. The equivalent of the meter authorized by the U. S. Government is 39.37 inches (exact value 39.370432 in.).

11. MEASURES OF SURFACE

100 square millimeters (mm.^{2})	= 1	square centimeter	sq. cm.
100 " centimeters	= 1	" decimeter	sq. dm.
100 " decimeters	= 1	" meter	sq. m.
100 " meters	= 1	" decameter	sq. dm.
100 " decameters	= 1	" hectometer	sq. hm.
100 " hectometers	= 1	" kilometer	sq. km.

In measuring land, the square meter is called a centare (ca.); the square decameter an are (a.); and the square hektometer a hektare (ha.)

12. MEASURES OF VOLUME

1000 cubic millimeters (mm.^{3})	= 1	cubic centimeter	cu. cm.
1000 " centimeters	= 1	" decimeter	" dm.
1000 " decimeters	= 1	" meter	" m.

13. . MEASURES OF CAPACITY

10 milliliters (Ml.)	= 1 centiliter	cl.
10 centiliters	= 1 deciliter	dl.
10 deciliters	= 1 liter	l.
10 liters	= 1 decaliter	dl.
10 decaliters	= 1 hectoliter	hl.
10 hectoliters	= 1 kiloliter	kl.

The liter is equal to the volume that is occupied by 1 cu. decimeter.

14. MEASURES OF WEIGHT

10 milligrams (Mg.)	= 1 centigram	cg.
10 centigrams	= 1 decigram	dg.
10 decigrams	= 1 gram	g.
10 grams	= 1 decagram	dg.
10 decagrams	= 1 hectogram	hg.
10 hectograms	= 1 kilogram	kg.
10 kilograms	= 1 myriagram	mg.
10 myriagrams	= 1 quintal	q.
10 quintals	= 1 tonneau (ton)	t.

The gram is the weight of 1 cubic centimeter of pure distilled water at a temperature of 39.2 deg. F. The kilogram is the weight of 1 liter of water. The tonneau is the weight of 1 cubic meter of water.

15. COMPARATIVE TABLE OF THE UNITED STATES AND METRIC SYSTEMS

U. S.		*Metric*	*Metric*		*U. S.*
1 grain	=	0.0648 g.	1 grain	=	15.433 gr.
1 pound (avp.)	=	0.4536 kg.	1 kilogram	=	2.2047 lb.
1 ton (2240 lb.)	=	1.0160 t.	1 tonneau	=	0.9843 T (2240 lb.)
1 ton (2000 lb.)	=	0.9071 t.	1 tonneau	=	1.1024 T (2000 lb.)
1 inch	=	25.400 mm.	1 millimeter	=	0.0394 in.
1 foot	=	0.3048 m.	1 meter	=	3.2807 ft.
1 mile	=	1.6094 km.	1 kilometer	=	0.6213 miles
1 square inch	=	645.2 sq. mm.	1 square millimeter	=	0.00155 sq. in.
1 square foot	=	0.0929 sq. m.	1 square meter	=	10.763 sq. ft.
1 acre	=	40.47 ares	1 are	=	0.02471 acres
1 square mile	=	2.59 sq. km.	1 square kilometer	=	0.3861 sq. mi.
1 cubic inch	=	16.39 cu. cm.	1 cubic centimeter	=	0.0610 cu. in.
1 cubic foot	=	0.0283 cu. cm.	1 cubic meter	=	35.31 cu. ft.
1 cubic yard	=	0.7646 cu. cm.	1 cubic meter	=	1.3078 cu. yd.
1 quart, dry	=	1.101 l.	1 liter	=	0.908 qt., dry
1 quart, liquid	=	0.9465 l.	1 liter	=	1.0566 qt., liquid

1 foot-pound	=	0.1383 kilogram-meters
1 kilogram-meter	=	7.2313 foot-pounds
1 pound per foot	=	1.488 kilogrammes per meter
1 kilogramme per meter	=	0.6720 pounds per foot
1000 lb. per sq. in.	=	0.703 kilogrammes per sq. mm.
1 kg. per sq. mm.	=	1422 lb. per sq. in.
1 lb. per sq. ft.	=	4.882 kg. per sq. m.
1 kg. per sq. mm.	=	0.2048 lb. per sq. ft.
1 lb. per cu. ft.	=	16.02 kg. per cu. m.
1 kg. per cu. mm.	=	0.0624 lb. per cu. ft.
1 deg. F.	=	0.5556 deg. C.
1 deg. C.	=	1.8 deg. F.

16. METRIC CONVERSION TABLES

To convert metric measurements into U. S. standard measurements multiply the metric quantity by the corresponding conversion figure as given in the following tables. The answer will be the equivalent U. S. standard measure.

17. MEASUREMENT OF LENGTH

	To Inches	*To Feet*	*To Yards*	*To Miles*
Millimeters	= 0.0393	0.0032	....	.....
Centimeters	= 0.3937	0.0327	0.0194	.....
Meters	= 39.37	3.28	1.09	.....
Kilometers	=	3280.87	1090.4	0.621

18. MEASURES OF SURFACE

		To Sq. In.	*To Sq. Ft.*
Square millimeters	=	0.0015	....
Square centimeters	=	0.1550	0.0010
Square meters	=	1550.03	10.76

Square kilometers	×	247.1	= acres
" "	×	0.3861	= sq. miles
Hectare	×	2.471	= acres

19. MEASURES OF VOLUME

Cubic centimeters	÷	16.383	= cu. in.
Cubic meters	×	35.315	= cu. ft.
Cubic meters	×	1.308	= cu. yd.
Cubic meters	×	264.2	= U. S. gallons (231 cu. in.)

20. MEASURES OF CAPACITY

Liter	×	0.908	= quarts
Liter	×	0.264	= U. S. gallon (231 cu. in.)
Liter	×	0.035	= cu. ft.
Liter	×	61.022	= cu. in.
Hectoliters	×	3.531	= cu. ft.
"	×	2.84	= U. S. struck bushels (2150.42 cu. in.)
"	×	1.31	= cu. yd.

24. MEASURES OF WEIGHT

		To Grains	*To Ounces*	*To Pounds*
Milligrams	×	0.0154	....	....
Grams	×	15.42	0.0352	....
Kilograms	×	....	35.3	2.204
Quintal	×	....	....	220.46
Tonneau	×	....	....	2204.62

Kilograms ÷ 907.2 = tons of 2000 lb.

25. MISCELLANEOUS CONVERSION FACTORS

Grams per cu. cm. ÷ 27.7	= lb. per cu. in.
Kilograms per sq. cm. × 14.223	= lb. per sq. in.
Kilogram-meters × 7.233	= lb. ft.
Kilograms per meter × 0.672	= lb. per ft.
Kilograms per cu. m. × 0.062	= lb. per cu. ft.
Kilograms per cheval × 2.235	= lb. per hp.
Calorie × 3.968	= B.t.u.
Cheval vapeur × 0.9863	= hp.
Gravity	= 980.94 cm. per sec.

26. MISCELLANEOUS EQUIVALENTS

Atmospheric pressure =
- 29.92 inches mercury at 32 deg. F. =
- 30 in. mercury at 62 deg. F. =
- 76 cm. mercury =
- 33.93 ft. water =
- 14.7 lb. per sq. in.

1 Horsepower =
- 0.7457 kw. =
- 1.014 cheval vapeur = 0.7604 poncelets =
- 0.7074 B.t.u. per sec. =
- 550 ft.-lb. per sec.

1 Cheval vapeur =
- 0.7355 kw. =
- 0.9863 hp. =
- 0.75 poncelets

1 Kilowatt =
- 1.341 hp. =
- 1.36 cheval vapeur =
- 1.02 poncelets.

1 British thermal unit (B.t.u.) =
- 777.5 ft.-lb. =
- 0.252 kilogram-calories

1 Kilogram-calorie =
- 3.968 B.t.u. =
- 3086 ft.-lb.

Joule × 0.7373 = ft.-lb.

Watts ÷ 746 = hp.

Watts × 0.7373 = ft. lb. per sec.

30. 1914 AMERICAN PIPE STANDARDS

(Table 30 on following page.)

Cast-Iron Pipe and Flanges for 125 lb. per sq. in. Working Pressure

Effective Jan. 1, 1914

The thickness of the pipe walls given in Table 30 for both the 125 and the 250 lb. per sq. in. pipes is calculated from the formula

$$t = \left[\frac{P \times 100}{4 \times S} D + 0.333 \left(1 - \frac{D}{100}\right)\right] 1.2$$

as recommended by The American Society of Mechanical Engineers and adopted as the American standard.

t = Thickness of pipe wall.
P = Pressure in lb. per sq. in.
D = Inside diameter of pipe in inches.
S = 1800 = Stress in pipe wall.

The value of $S = \dfrac{r \times P}{t}$ where r = inside radius of pipe in inches.

Bolt holes are drilled straddle center line of flange.

30. Cast-Iron Pipe and Flanges for 125 lb. per sq. in. Working Pressure

Dimensions of Pipe in Inches			Stress in Pipe per Sq. In.	Dimensions of Flange in Inches			No. and Dimensions of Bolts in Inches				Diameter Bolt Holes	Stress per Sq. In. on Bolt Metal
Inside Diameter	Thickness t	Minimum Thickness Fractions of an Inch		Diameter	Thickness	Bolt Circle	Number	Diameter	Length	Length of Studs With Two Nuts		
1	0.43	7/16	143	4	7/16	3	4	7/16	1½	....	9/16	264
1¼	0.44	7/16	178	4½	½	3⅜	4	7/16	1½	....	9/16	412
1½	0.45	7/16	214	5	9/16	3⅞	4	½	1¾	....	⅝	438
2	0.46	7/16	286	6	⅝	4¾	4	⅝	2	....	¾	486
2½	0.48	7/16	357	7	11/16	5½	4	⅝	2¼	....	¾	750
3	0.50	7/16	428	7½	¾	6	4	⅝	2¼	....	¾	1093
3½	0.52	7/16	500	8½	13/16	7	4	⅝	2½	....	¾	1488
4	0.53	½	500	9	15/16	7½	8	⅝	2¾	....	¾	972
4½	0.55	½	562	9¼	15/16	7¾	8	¾	2¾	....	⅞	823
5	0.56	½	625	10	15/16	8½	8	¾	2¾	....	⅞	1016
6	0.60	9/16	667	11	1	9½	8	¾	3	....	⅞	1463
7	0.63	⅝	700	12½	1 1/16	10¾	8	¾	3	....	⅞	1991
8	0.66	⅝	800	13½	1⅛	11¾	8	¾	3¼	....	⅞	2600
9	0.70	11/16	818	15	1⅛	13¼	12	¾	3¼	....	⅞	2194
10	0.73	¾	833	16	1 3/16	14¼	12	⅞	3½	....	1	1948

Continued on following pages.

30. Cast-Iron Pipe and Flanges for 125 lb. per sq. in. Working Pressure—Continued

Dimensions of Pipe in Inches			Stress in Pipe per Sq. In.	Dimensions of Flange in Inches			No. and Dimensions of Bolts in Inches				Diameter Bolt Holes	Stress per Sq. In. on Bolt Metal
Inside Diameter	Thickness t	Minimum Thickness Fractions of an Inch		Diameter	Thickness	Bolt Circle	Number	Diameter	Length	Length of Studs With Two Nuts		
12	0.80	13/16	923	19	1 1/4	17	12	7/8	3 1/2	....	1	2805
14	0.86	7/8	1000	21	1 3/8	18 3/4	12	1	4	....	1 1/8	2915
15	0.90	7/8	1072	22 1/4	1 3/8	20	16	1	4	....	1 1/8	2510
16	0.93	1	1000	23 1/2	1 7/16	21 1/4	16	1	4	....	1 1/8	2856
18	1.00	1 1/16	1059	25	1 9/16	22 3/4	16	1 1/8	4 1/2	....	1 1/4	2865
20	1.07	1 1/8	1111	27 1/2	1 11/16	25	20	1 1/8	4 3/4	....	1 1/4	2829
22	1.13	1 3/16	1158	29 1/2	1 13/16	27 1/4	20	1 1/4	5	....	1 3/8	2660
24	1.20	1 1/4	1200	32	1 7/8	29 1/2	20	1 1/4	5 1/4	....	1 3/8	3166
26	1.27	1 5/16	1238	34 1/4	2	31 3/4	24	1 1/4	5 1/2	....	1 3/8	3096
28	1.33	1 3/8	1273	36 1/2	2 1/16	34	28	1 1/4	5 1/2	....	1 3/8	3078
30	1.40	1 7/16	1304	38 3/4	2 1/8	36	28	1 3/8	5 3/4	....	1 1/2	2985
32	1.47	1 1/2	1333	41 3/4	2 1/4	38 1/2	28	1 1/2	6 1/4	....	1 5/8	2775
34	1.54	1 9/16	1360	43 3/4	2 5/16	40 1/2	32	1 1/2	6 1/2	....	1 5/8	2741
36	1.60	1 5/8	1385	46	2 3/8	42 3/4	32	1 1/2	6 1/2	....	1 5/8	3073
38	1.67	1 11/16	1407	48 3/4	2 3/8	45 1/4	32	1 5/8	6 3/4	9	1 3/4	2924

30. Cast-Iron Pipe and Flanges for 125 lb. per sq. in. Working Pressure—Continued

Dimensions of Pipe in Inches			Stress in Pipe per Sq. In.	Dimensions of Flange in Inches			No. and Dimensions of Bolts in Inches				Diameter Bolt Holes	Stress per Sq. In. on Bolt Metal
Inside Diameter	Thickness t	Minimum Thickness Fractions of an Inch		Diameter	Thickness	Bolt Circle	Number	Diameter	Length	Length of Studs With Two Nuts		
40	1.73	1¾	1428	50¾	2½	47¼	36	1⅝	7	9	1¾	2880
42	1.82	1 13/16	1448	53	2⅝	49½	36	1⅝	7¼	9½	1¾	3175
44	1.87	1⅞	1467	55¼	2⅝	51¾	40	1⅝	7¼	9½	1¾	3136
46	1.94	1 15/16	1484	57¼	2 11/16	53¾	40	1⅝	7½	9½	1¾	3428
48	2.00	2	1500	59½	2¾	56	44	1⅝	7½	9½	1¾	3393
50	2.07	2 1/16	1515	61¾	2¾	58¼	44	1¾	7¾	10	1⅞	3195
52	2.14	2⅛	1530	64	2⅞	60½	44	1¾	8	10½	1⅞	3456
54	2.20	2 3/16	1543	66¼	3	62¾	44	1¾	8¼	10½	1⅞	3726
56	2.27	2¼	1555	68¾	3	65	48	1¾	8¼	10½	1⅞	3674
58	2.34	2 5/16	1567	71	3⅛	67¼	48	1¾	8½	11	1⅞	3941
60	2.41	2 7/16	1538	73	3⅛	69¼	52	1¾	8½	11	1⅞	3892
62	2.47	2½	1550	75¾	3¼	71¾	52	1⅞	9	11½	2	3538
64	2.54	2 9/16	1561	78	3¼	74	52	1⅞	9	11½	2	3770
66	2.61	2⅝	1572	80	3⅜	76	52	1⅞	9¼	11½	2	4010
68	2.68	2 11/16	1582	82¼	3⅜	78¼	56	1⅞	9¼	11½	2	3952

Concluded on following page.

30. Cast-Iron Pipe and Flanges for 125 lb. per sq. in. Working Pressure—Concluded

Dimensions of Pipe in Inches			Stress in Pipe per Sq. In.	Dimensions of Flange in Inches			No. and Dimensions of Bolts in Inches				Diameter Bolt Holes	Stress per Sq. In. on Bolt Metal
Inside Diameter	Thickness t	Minimum Thickness Fractions of an Inch		Diameter	Thickness	Bolt Circle	Number	Diameter	Length	Length of Studs With Two Nuts		
70	2.74	2¾	1591	84½	3½	80½	56	1⅞	9½	12	2	4188
72	2.81	2 13/16	1600	86½	3½	82½	60	1⅞	9½	12	2	4136
74	2.88	2⅞	1609	88½	3⅝	84½	60	1⅞	9¾	12	2	4368
76	2.94	2 15/16	1617	90¾	3⅝	86½	60	1⅞	9¾	12	2	4608
78	3.01	3	1625	93	3¾	88¾	60	2	10	12½	2⅛	4325
80	3.08	3 1/16	1633	95¼	3¾	91	60	2	10	12½	2⅛	4549
82	3.15	3⅛	1640	97½	3⅞	93¼	60	2	10½	13	2⅛	4779
84	3.21	3 3/16	1647	99¾	3⅞	95½	64	2	10½	13	2⅛	4702
86	3.28	3¼	1653	102	4	97¾	64	2	10½	13	2⅛	4928
88	3.35	3 5/16	1660	104¼	4	100	68	2	10½	13	2⅛	4857
90	3.41	3⅜	1667	106½	4⅛	102¼	68	2⅛	11	14	2¼	4416
92	3.48	3½	1643	108¾	4⅛	104½	68	2⅛	11	14	2¼	4615
94	3.55	3 9/16	1649	111	4¼	106¼	68	2⅛	11¼	14	2¼	4817
96	3.62	3⅝	1655	113¼	4¼	108½	68	2¼	11½	14½	2⅜	4401
98	3.68	3 11/16	1661	115½	4⅜	110¾	68	2¼	11½	14½	2⅜	4587
100	3.75	3¾	1667	117¾	4⅜	113	68	2¼	11½	14½	2⅜	4776

Continued on following pages:

31. Cast-Iron Pipe and Flanges for 250 lb. per sq. in. Working Pressure

Dimensions of Pipe in Inches			Stress in Pipe per Sq. In.	Dimensions of Flange in Inches			Number and Dimensions of Bolts in Inches				Diameter Bolt Holes	Stress per Sq. In. on Bolt Metal
Inside Diameter	Thickness t	Minimum Thickness Fractions of an Inch		Diameter	Thickness	Diameter of Bolt Circle	Number	Diameter	Length of Bolts	Length of Studs With Two Nuts		
1	0.45	1/2	250	4 1/2	11/16	3 1/4	4	1/2	2	....	5/8	389
1 1/4	0.47	1/2	312	5	3/4	3 3/4	4	1/2	2 1/4	....	5/8	609
1 1/2	0.49	1/2	375	6	13/16	4 1/2	4	5/8	2 1/2	....	3/4	547
2	0.51	1/2	500	6 1/2	7/8	5	4	5/8	2 1/2	....	3/4	972
2 1/2	0.53	9/16	555	7 1/2	1	5 7/8	4	3/4	3	....	7/8	1016
3	0.56	9/16	667	8 1/4	1 1/8	6 5/8	8	3/4	3 1/4	....	7/8	731
3 1/2	0.59	9/16	778	9	1 3/16	7 1/4	8	3/4	3 1/4	....	7/8	995
4	0.61	5/8	800	10	1 1/4	7 7/8	8	3/4	3 1/2	....	7/8	1300
4 1/2	0.64	5/8	900	10 1/2	1 5/16	8 1/2	8	3/4	3 1/2	....	7/8	1646
5	0.67	11/16	909	11	1 3/8	9 1/4	8	3/4	3 3/4	....	7/8	2032
6	0.72	3/4	1000	12 1/2	1 7/16	10 5/8	12	3/4	3 3/4	....	7/8	1950
7	0.78	13/16	1077	14	1 1/2	11 7/8	12	7/8	4	....	1	1909
8	0.83	13/16	1230	15	1 5/8	13	12	7/8	4 1/4	....	1	2493
9	0.89	7/8	1285	16 1/4	1 3/4	14	12	1	4 3/4	....	1 1/8	2410
10	0.94	15/16	1333	17 1/2	1 7/8	15 1/4	16	1	5	....	1 1/8	2231
12	1.05	1	1500	20 1/2	2	17 3/4	16	1 1/8	5 1/4	....	1 1/4	2546
14	1.16	1 1/8	1555	23	2 1/8	20 1/4	20	1 1/8	5 1/2	....	1 1/4	2773
15	1.21	1 3/16	1579	24 1/2	2 3/16	21 1/2	20	1 1/4	5 3/4	....	1 3/8	2473

Concluded on following page:

31. Cast-Iron Pipe and Flanges for 250 lb. per sq. in. Working Pressure—Concluded

Dimensions of Pipe in Inches			Stress in Pipe per Sq. In.	Dimensions of Flange in Inches			Number and Dimensions of Bolts in Inches				Diameter Bolt Holes	Stress Sq. In. on Bolt Metal
Inside Diameter	Thickness t	Minimum Thickness Fractions of an Inch		Diameter	Thickness	Diameter of Bolt Circle	Number	Diameter	Length of Bolts	Length of Studs With Two Nuts		
16	1.27	1¼	1600	25½	2¼	22½	20	1¼	6	....	1⅜	2814
18	1.37	1⅜	1636	28	2⅜	24¾	24	1¼	6¼	....	1⅜	2968
20	1.48	1½	1666	30½	2½	27	24	1⅜	6½	....	1½	3096
22	1.59	1 9/16	1760	33	2⅝	29¼	24	1½	7	....	1⅝	3058
24	1.70	1⅝	1846	36	2¾	32	24	1⅝	7½	9½	1¾	3110
26	1.81	1 13/16	1793	38¼	2 13/16	34½	28	1⅝	7¾	10	1¾	3126
28	1.91	1⅞	1866	40¾	2 15/16	37	28	1⅝	8	10	1¾	3629
30	2.02	2	1875	43	3	39¼	28	1¾	8¼	10½	1⅞	3615
32	2.13	2⅛	1882	45¼	3⅛	41½	28	1⅞	8½	11	2	3501
34	2.24	2¼	1889	47½	3¼	43½	28	1⅞	9	11½	2	3952
36	2.35	2⅜	1894	50	3⅜	46	32	1⅞	9¼	11½	2	3877
38	2.46	2 7/16	1948	52¼	3 7/16	48	32	1⅞	9¼	11½	2	4320
40	2.56	2 9/16	1953	54½	3 9/16	50¼	36	1⅞	9½	12	2	4255
42	2.67	2 11/16	1953	57	3 11/16	52¾	36	1⅞	9¾	12	2	4691
44	2.78	2 13/16	1955	59¼	3¾	55	36	2	10	12½	2⅛	4587
46	2.89	2⅞	2000	61½	3⅞	57¼	40	2	10¼	13	2⅛	4512
48	3.00	3	2000	65	4	60¾	40	2	10½	13	2⅛	4913

FIRE PROTECTION

40. When making provisions for fire protection it becomes necessary to calculate the effectiveness of a stream of water when led through a given length of hose for a given pressure at the hydrant or pump, or to find the pressure required to throw a stream of water a given height or a given distance.

41. To assist in making calculations of this nature we present by permission of the National Board of Fire Underwriters the following data based on tests made by their engineers with the assistance of the New York Fire Department. With these data any one can calculate the quantity of water required, the hydrant or pump pressure necessary for one or more effective fire streams when using a given size nozzle. Contractors will also find these tables useful in calculating their pumping requirements for removing earth, etc., by means of hydraulic jets.

42. Example: What would be the required capacity and discharge pressure for a pump to maintain a nozzle pressure of 100 lb. per sq. in. on two 1⅛-in. nozzles, through 300 ft. of 2½-in. hose? From table, par. 61 we find that gal. per min. discharged by a 1⅛-in. nozzle with 100 lb. per sq. in. nozzle pressure is 374. Then for two nozzles we will require $2 \times 374 = 748$ gal. per min. From table, par. 61 we find the hydrant pressure necessary to maintain 100-lb. per sq. in. nozzle pressure through 300 ft. of 2½-in. hose is 201 lb. per sq. in. As we intend to attach the hose direct to the pump the hydrant pressure will correspond to the discharge pressure at the pump, so we select a pump to have a capacity of 748 gal. per min. against 201-lb. per sq. in. discharge pressure.

43. When it is necessary to locate a pump some distance away from the beginning of the hose line and at a lower elevation, it is necessary to add the pipe friction and the static head to the hydrant pressure in order to obtain the correct pump discharge pressure.

FIRE STREAM TABLES

44. These tables are arranged to show the pressures required at the hydrant or fire engine, while stream is flowing, to maintain nozzle pressures given in the first columns, through various lengths of 2½-, 3- and 3½-in. rubber-lined hose in single lines and two lines of 2½-in. hose siamesed.

45. Nozzle pressures of 40 to 60 lb. per sq. in. from 1⅛- and 1¼-in. nozzles will give streams which may be classed as good and which can be handled without special appliances; for deluge sets, turret pipes, etc., with 1½-in. and larger nozzles, 60 to 90 lb. per sq. in. nozzle pressure is desirable for effective fire fighting; the height, area and general character of the building are factors in determining at what pressure a stream may be considered good, as well as in determining whether a nozzle is of sufficient size to furnish an effective stream, nothing less than 1⅛-in. being considered as effective for outside work, except for fires in small buildings. In this connection it should be noted that a 1- or 1⅛-in. ring tip delivers a stream about ⅛-in. smaller than the diameter of the tip.

46. The pressure at the hydrant or fire engine is that indicated by a gage attached to the hydrant or fire engine while the stream is flowing. The pressure at the nozzle is that indicated by a pitot gage held in the stream.

47. The hydrant (or engine) pressures are obtained by adding to the nozzle pressure the friction loss in the hose, and also the small additional loss in the hydrant outlet or engine discharge.

48. Friction losses in hose are based on tests of best quality rubber-lined fire hose and are for 100-ft. lengths measured without pressure applied. Diameters of hose, as measured under 75 lb. per sq. in. pressure, assumed as the average working condition, were as follows: For nominal 2½-in., 2.575 or about 2$\frac{9}{16}$ in.; for nominal 3-in., 3.125 or 3⅛ in.; for nominal 3½-in., 3.685 or about 3$\frac{11}{16}$ in.

49. The smoothness of the lining has a very considerable effect on the friction loss, some samples tested showing losses 50 per cent. in excess of those given. A slight variation in diameter also

produces a marked difference in friction loss; in the case of 2½-in. hose, a variation of 1/16 in. in diameter will result in 10 per cent. difference in loss. If properly beveled 2½-in. couplings are used on 3-in. hose, the loss of pressure due to them will be less than 5 per cent. of that gained by the use of the larger hose. For instance, for a flow of 300 gal. per min., the loss in 2½-in. hose will be about 21 lb. per sq. in.; in 3-in. hose with 3-in. couplings about 8 lb. per sq. in.; and in 3-in. hose with 2½-in. couplings about 8½ lb. per sq. in.

50. For siamesed lines, an allowance was made for the loss in the siamese connection and for 20 ft. of 3½-in. lead hose.

51. The pressures given are for the nozzle at the same elevation as the hydrant or engine discharge outlet. Add or subtract 1 lb. per sq. in. to the pressure given for each 2⅓ ft. difference in elevation. The arrangement of table, par. 58 allows a comparison to be readily made of the results obtainable with 3-in. hose and siamesed lines against single lines of 2½-in. hose.

52. An approximate formula for the friction loss in 2½-in. hose is as follows:

53. Pressure loss in each hundred feet, in pounds per sq. in. = $2\,Q^2 + Q$, where Q is the quantity in gallons divide by 100. (For less than 100 gal. per min. pressure loss $= 2\,Q^2 + \frac{1}{2}Q$.)

54. The approximate figures for the friction loss in other sizes for the same quantity flowing can be obtained by dividing the friction loss in 2½-in. hose by the following factors:

SINGLE LINES

2¾ in.	3 in.	3½ in.	4 in.	4½ in.	5 in.
1.66	2.6	5.8	11.0	19.5	32.0

SIAMESED LINES OF EQUAL LENGTH

2-2½ in.	3-2½ in.	2-3 in.	3-3 in.	1-3 in. & 1-2½ in.
3.6	7.75	9.35	20.4	6.1

TABLE OF NOZZLE FACTORS

55. For use in obtaining discharge from smooth nozzle larger than those given in tables when nozzle pressure is obtained with a Pitot gage.

The discharge in gallons per minute is equal to the square root of the pressure multiplied by the factor.

Diameter of the nozzle in inches	Factors	
	For Fresh Water	For Salt (sea) Water
2	118.96	117.45
2¼	150.56	148.64
2½	185.88	183.50
2¾	224.91	222.05
3	267.66	264.25
3¼	314.13	310.13
3½	364.32	359.68
3¾	418.23	412.90
4	475.85	469.79
4¼	537.19	530.35
4½	602.25	594.58
4¾	671.02	662.48
5	743.51	734.03
6	1,070.64	1,057.00

For any size nozzles, the discharge, for fresh water, can be determined by the following formula:

Gallons per minute $= 29.83\, c\, d^2 \sqrt{p}$.

Where d = diameter of nozzle in inches, measured to 1/1000 of an inch.

p = pressure recorded on Pitot gage in pounds.

c = a constant, varying from 0.990 for 1-inch nozzle to 0.997 for 6-inch nozzle.

For ordinary use, the formula can be reduced to:

Gallons per minute $= 29.7\, d^2 \sqrt{p}$.

FORMULA FOR OBTAINING APPROXIMATE NOZZLE OR ENGINE PRESSURES, LENGTH OF LINE AND SIZE OF NOZZLE BEING GIVEN

56. Nozzle Pressure in pounds $= \dfrac{\text{Engine Pressure}}{1.1 + KL}$

Engine Pressure in pounds = Nozzle Pressure $(1.1 + KL)$
L = Number of 50-foot lengths of hose.
K = Constant, varying with size of nozzle and hose. See Table following:

Size Nozzle, Inches	*K* for					
	Single Line 2½ in. Hose	Single Line 3 in. Hose	Single Line 3½ in. Hose	Two 2½ in. Lines Siamesed *	Two 3 in. Lines Siamesed *	3 Lines 2½ in. Hose *
1	0.105	0.038	......	0.025	......	......
1⅛	0.167	0.062	......	0.043	......	......
1¼	0.248	0.092	0.039	0.066	0.023	0.028
1⅜	0.341	0.137	0.059	0.096	0.034	0.043
1½	0.505	0.192	0.084	0.135	0.051	0.061
1⅝	0.680	0.266	0.113	0.184	0.068	0.084
1¾	0.907	0.351	0.152	0.242	0.093	0.115
2	1.550	0.605	0.250	0.418	0.157	0.190

*Allowance is made for loss in deluge set; these values will also give approximately correct figures for turret nozzles and water tower, except that in the latter, pressure equal to 0.434 times the height of tower must be subtracted from the engine pressure, before solving for nozzle pressure.

57. EFFECTIVE REACH OF FIRE STREAMS

Showing the Distance in Feet from the Nozzle at which Streams will do Effective Work with a Moderate Wind Blowing. With a Strong Wind the Reach is Greatly Reduced.

Pressure at Nozzle lb. per sq. in.	Size of Nozzle									
	1-In.		1⅛-In.		1¼-In.		1⅜-In.		1½-In.	
	Vertical Distance, Feet	Horizontal Distance, Feet	Vertical Distance, Feet	Horizontal Distance, Feet	Vertical Distance, Feet	Horizontal Distance, Feet	Vertical Distance, Feet	Horizontal Distance, Feet	Vertical Distance, Feet	Horizontal Distance, Feet
20	35	37	36	38	36	39	36	40	37	42
25	43	42	44	44	45	46	45	47	46	49
30	51	47	52	50	52	52	53	54	54	56
35	58	51	59	54	59	58	60	59	62	62
40	64	55	65	59	65	62	66	64	69	66
45	69	58	70	63	70	66	72	68	74	71
50	73	61	75	66	75	69	77	72	79	75
55	76	64	79	69	80	72	81	75	83	78
60	79	67	83	72	84	75	85	77	87	80
65	82	70	86	75	87	78	88	79	90	82
70	85	72	88	77	90	80	91	82	92	84
75	87	74	90	79	92	82	93	84	94	86
80	89	76	92	81	94	84	95	86	96	88
85	91	78	94	83	96	87	97	88	98	90
90	92	80	96	85	98	89	99	90	100	91

Note.—Nozzle pressures are as indicated by Pitot tube. The horizontal and vertical distances are based on experiments by Mr. John R. Freeman, "Transactions," Am. Soc. C. E., Vol. XXI.

58. FRICTION LOSS IN FIRE HOSE

Based on Tests of Best Quality Rubber-Lined Fire Hose*

Flow, Gallons per Minute	Pressure Loss in Each 100 Ft. of Hose, Lb. per sq. in.				Flow, Gallons per Minute	Pressure Loss in Each 100 Ft. of Hose, Lb. per sq. in.		
	2½-in. Hose	3-in. Hose	3½-in. Hose	2 Lines of 2½-in. Siamesed		3-in. Hose	3½-in. Hose	2 Lines of 2½-in. Siamesed
140	5.2	2.0	0.9	1.4	525	23.2	10.5	16.6
160	6.6	2.6	1.2	1.9	550	25.2	11.4	18.1
180	8.3	3.2	1.5	2.3	575	27.5	12.4	19.0
200	10.1	3.9	1.8	2.8	600	29.9	13.4	21.2
220	12.0	4.2	2.1	3.3	625	32.0	14.4	23.0
240	14.1	5.4	2.5	3.9	650	34.5	15.5	24.8
260	16.4	6.3	2.9	4.5	675	37.0	16.6	26.5
280	18.7	7.2	3.3	5.2	700	39.5	17.7	28.3
300	21.2	8.2	3.7	5.9	725	42.3	18.9	30.2
320	23.8	9.3	4.2	6.6	750	45.0	20.1	32.2
340	26.9	10.5	4.7	7.4	775	47.8	21.4	34.2
360	30.0	11.5	5.2	8.3	800	50.5	22.7	36.2
380	33.0	12.8	5.8	9.2	825	53.5	24.0	38.4
400	36.2	14.1	6.3	10.1	850	56.5	25.4	40.7
425	40.8	15.7	7.0	11.3	875	59.7	26.8	43.1
450	45.2	17.5	7.9	12.5	900	63.0	28.2	45.2
475	50.0	19.3	8.7	13.8	1,000	76.5	34.3	55.0
500	55.0	21.2	9.5	15.2	1,100	91.5	41.0	65.5

*Rough rubber lining is liable to increase the losses given in the table as much as 50 per cent

59. DISCHARGE TABLE FOR SMOOTH NOZZLES

NOZZLE PRESSURE MEASURED BY PITOT GAGE

Nozzle Pressure in lb. per sq. in.	Nozzle Diam. in Inches 1	1⅛	1¼	1⅜	1½	Nozzle Pressure in lb. per sq. in.	Nozzle Diam. in Inches 1	1⅛	1¼	1⅜	1½
	Gallons per Minute						Gallons per Minute				
5	66	84	103	125	149	60	229	290	357	434	517
6	72	92	113	137	163	62	233	295	363	441	525
7	78	99	122	148	176	64	237	299	369	448	533
8	84	106	131	158	188	66	240	304	375	455	542
9	89	112	139	168	200	68	244	308	381	462	550
10	93	118	146	177	211	70	247	313	386	469	558
12	102	130	160	194	231	72	251	318	391	475	566
14	110	140	173	210	249	74	254	322	397	482	574
16	118	150	185	224	267	76	258	326	402	488	582
18	125	159	196	237	283	78	261	330	407	494	589
20	132	167	206	250	298	80	264	335	413	500	596
22	139	175	216	263	313	82	268	339	418	507	604
24	145	183	226	275	327	84	271	343	423	513	611
26	151	191	235	286	340	86	274	347	428	519	618
28	157	198	244	297	353	88	277	351	433	525	626
30	162	205	253	307	365	90	280	355	438	531	633
32	167	212	261	317	377	92	283	359	443	537	640
34	172	218	269	327	389	94	286	363	447	543	647
36	177	224	277	336	400	96	289	367	452	549	654
38	182	231	285	345	411	98	292	370	456	554	660
40	187	237	292	354	422	100	295	374	461	560	667
42	192	243	299	363	432	105	303	383	473	574	683
44	196	248	306	372	442	110	310	392	484	588	699
46	200	254	313	380	452	115	317	401	495	600	715
48	205	259	320	388	462	120	324	410	505	613	730
50	209	265	326	396	472	125	331	418	516	626	745
52	213	270	333	404	481	130	337	427	526	638	760
54	217	275	339	412	490	135	343	435	536	650	775
56	221	280	345	419	499	140	350	443	546	662	789
58	225	285	351	426	508	145	356	450	556	674	803
60	229	290	357	434	517	150	362	458	565	686	817

Continued on following page.

Assumed coefficient of discharge = 0.99, 0.99, 0.99, 0.9925, 0.995.

NOTE.—Coefficients of discharge are based on experiments by Mr. John R. Freeman. Transactions, Am. Soc. C. E., Vols. XXI and XXIV.

59. DISCHARGE TABLE FOR SMOOTH NOZZLES—Cont.

Nozzle Pressure in lb. per sq. in.	Nozzle Diam. in Inches 1⅝	1¾	1⅞	2	2¼	Nozzle Pressure in lb. per sq. in.	Nozzle Diam. in Inches 1⅝	1¾	1⅞	2	2¼
	Gallons per Minute						Gallons per Minute				
5	175	203	234	266	337	60	607	704	810	920	1167
6	192	223	256	292	369	62	617	716	823	936	1187
7	207	241	277	315	399	64	627	727	836	951	1206
8	222	257	296	336	427	66	636	738	850	965	1224
9	235	273	314	357	452	68	646	750	862	980	1242
10	248	288	330	376	477	70	655	761	875	994	1260
12	271	315	362	412	522	72	665	771	887	1008	1278
14	293	340	391	445	564	74	674	782	900	1023	1296
16	313	364	418	475	603	76	683	792	911	1036	1313
18	332	386	444	504	640	78	692	803	924	1050	1330
20	350	407	468	532	674	80	700	813	935	1063	1347
22	367	427	490	557	707	82	709	823	946	1076	1364
24	384	446	512	582	739	84	718	833	959	1089	1380
26	400	464	533	606	769	86	726	843	970	1102	1396
28	415	481	554	629	799	88	735	853	981	1115	1412
30	429	498	572	651	826	90	743	862	992	1128	1429
32	443	514	591	673	854	92	751	872	1002	1140	1445
34	457	530	610	693	880	94	759	881	1012	1152	1460
36	470	546	627	713	905	96	767	890	1022	1164	1476
38	483	561	645	733	930	98	775	900	1032	1176	1491
40	496	575	661	752	954	100	783	909	1043	1189	1506
42	508	589	678	770	978	105	803	932	1070	1218	1542
44	520	603	694	788	1000	110	822	954	1095	1247	1579
46	531	617	710	806	1021	115	840	975	1120	1275	1615
48	543	630	725	824	1043	120	858	996	1144	1303	1649
50	554	643	740	841	1065	125	876	1016	1168	1329	1683
52	565	656	754	857	1087	130	893	1036	1191	1356	1717
54	576	668	769	873	1108	135	910	1056	1213	1382	1750
56	586	680	782	889	1129	140	927	1076	1235	1407	1780
58	596	692	796	905	1149	145	944	1095	1257	1432	1812
60	607	704	810	920	1168	150	960	1114	1279	1456	1843

Assumed coefficient of discharge = 0.995, 0.995, 0.996, 0.997, 0.997.

60. 1-IN. SMOOTH NOZZLE—2½- AND 3-IN. HOSE

Pressures Required at Hydrant or Fire Engine, while stream is flowing, to maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality 2½- and 3-in. Rubber-Lined Hose.

Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.	Discharge, Gal. per Min.	Single 2½-in. Lines										Single 3-in. Lines				Two 2½-in. Lines Siamesed			Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
		100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	700 ft.	800 ft.	1,000 ft.	1,200 ft.	800 ft.	1,000 ft.	1,200 ft.	1,500 ft.	1,000 ft.	1,500 ft.	2,000 ft.	
20	132	25	30	35	39	44	49	53	58	68	77	35	39	42	48	33	40	46	20
25	148	31	37	43	49	55	60	66	72	84	95	43	48	52	59	41	49	57	25
30	162	38	44	51	58	65	72	78	85	99	112	52	57	62	70	49	59	68	30
35	175	44	52	59	67	75	83	91	98	114	130	60	66	72	81	57	68	79	35
40	187	50	59	68	77	86	94	103	112	130	148	68	75	82	92	65	78	90	40
45	198	56	66	76	86	96	106	115	125	145	165	77	84	92	103	72	86	99	45
50	209	62	73	84	95	106	117	128	139	160	182	85	93	102	114	80	95	110	50
55	219	68	80	92	104	116	128	140	152	175	199	93	102	112	125	88	105	121	55
60	229	75	88	101	114	127	140	153	166	192	218	102	112	122	137	96	114	132	60
65	238	81	95	109	123	137	151	165	179	207	235	110	121	131	148	103	122	141	65
70	247	87	102	117	132	147	162	177	192	222	252	118	130	141	159	111	132	152	70
75	256	93	109	125	141	157	173	189	205	237	269	127	139	151	170	120	142	164	75
80	264	99	116	133	150	167	183	200	217	251	285	135	148	161	181	128	151	175	80
85	272	105	123	141	159	177	195	212	230	266	302	143	156	170	191	135	159	184	85
90	280	111	130	149	167	186	205	224	243	280	...	151	165	180	202	143	169	195	90
95	287	117	137	157	177	196	216	236	256	295	...	158	173	189	211	150	177	204	95
100	295	123	144	165	185	206	227	247	268	310	...	167	183	199	223	157	186	215	100

61. 1⅛-IN. SMOOTH NOZZLE—2½- AND 3-IN. HOSE

Pressures Required at Hydrant or Fire Engine, while stream is flowing, to maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality 2½- and 3-in. Rubber-Lined Hose

Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.	Discharge, Gal. per Min.	Single 2½-in. Lines										Single 3-in. Lines							Two 2½-in. Lines Siamesed					Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
		100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	700 ft.	800 ft.	1,000 ft.	1,200 ft.	400 ft.	600 ft.	800 ft	1,000 ft.	1,200 ft.	1,500 ft.	1,800 ft.	800 ft.	1,000 ft.	1,200 ft.	1,500 ft.	1,800 ft.	
20	167	28	35	42	49	56	64	71	78	92	107	32	37	43	48	54	62	71	38	42	46	53	60	20
25	187	35	44	53	62	71	79	88	97	115	133	40	46	53	60	67	77	87	45	50	55	63	70	25
30	205	42	52	63	73	84	95	105	116	137	158	47	55	63	71	79	91	103	53	59	65	74	82	30
35	221	49	61	73	85	97	110	122	134	158	183	55	65	74	83	93	107	121	62	69	76	86	96	35
40	237	55	69	83	96	110	124	138	151	179	206	63	73	84	95	105	121	137	70	78	86	97	108	40
45	251	62	77	93	108	123	139	154	169	200	230	70	82	94	106	118	135	153	79	87	95	108	121	45
50	265	69	86	103	120	137	154	171	188	222	256	78	91	104	117	130	150	169	88	98	107	121	135	50
55	277	76	94	112	131	149	168	186	204	241	278	86	100	114	128	142	164	185	96	107	117	132	147	55
60	290	83	103	123	143	163	183	203	223	263	304	93	109	124	139	155	178	201	105	116	127	143	160	60
65	301	89	111	132	154	175	197	218	240	283	326	101	117	134	151	167	192	217	114	126	138	156	174	65
70	313	96	119	142	165	188	211	234	257	303	....	108	126	144	162	180	206	233	122	135	148	167	186	70
75	324	103	128	152	177	202	227	252	276	325	....	116	135	154	173	192	221	249	130	144	157	178	198	75
80	335	110	136	162	188	215	241	267	294	....	....	124	144	165	185	206	236	267	138	153	167	189	210	80
85	345	116	144	171	199	226	254	282	309	....	....	131	153	174	195	217	249	281	147	163	178	201	224	85
90	355	123	152	181	210	240	269	298	327	....	....	139	161	184	207	229	263	297	156	172	188	212	237	90
95	365	130	160	191	222	252	283	314	....	....	....	146	170	194	218	242	277	313	164	181	198	224	249	95
100	374	136	168	201	233	265	297	329	....	....	....	154	178	203	228	253	290	....	172	190	208	235	261	100

62. 1¼-IN. SMOOTH NOZZLE—2½- AND 3-IN. HOSE

Nozzle Pressure Indicated by Pitot Gage. lb. per sq. in.	Discharge, Gal. per Min.	Pressures Required at Hydrant or Fire-Engine, while Stream is Flowing, to Maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality 2½- and 3-in. Rubber-Lined Hose																							Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
		Single 2½-in. Lines										Single 3-in. Lines							Two 2½-in. Lines Siamesed						
		100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	700 ft.	800 ft.	1,000 ft.	1,200 ft.	400 ft.	600 ft.	800 ft.	1,000 ft.	1,200 ft.	1,500 ft.	1,800 ft.	600 ft.	800 ft.	1,000 ft.	1,200 ft.	1,500 ft.	1,800 ft.	
20	206	32	42	53	64	75	85	96	107	128	149	37	46	54	62	70	83	95	39	45	51	57	67	76	20
25	230	40	53	66	79	92	105	118	131	158	184	47	57	67	77	87	102	117	48	55	62	70	80	91	25
30	253	48	63	79	95	110	126	142	157	189	220	56	68	81	93	105	123	142	57	66	74	83	96	109	30
35	273	55	73	91	109	127	145	163	181	217	253	65	79	92	106	120	141	161	66	76	86	95	110	125	35
40	292	63	83	104	124	144	165	185	206	246	287	74	89	105	120	136	159	183	75	87	99	110	127	144	40
45	309	70	93	116	138	161	183	206	229	274	319	83	100	117	135	152	178	204	84	96	109	121	140	158	45
50	326	78	103	128	153	178	203	228	253	303	...	91	111	130	149	168	197	226	93	107	121	135	155	176	50
55	342	86	113	140	167	191	222	249	276	330	...	100	121	142	163	184	216	247	102	117	132	147	169	192	55
60	357	93	123	152	182	211	241	270	300	...	...	109	132	155	178	201	235	270	111	128	144	160	185	210	60
65	372	101	133	164	196	228	260	292	323	...	...	118	143	167	192	217	254	291	120	137	155	173	199	225	65
70	386	108	142	176	210	244	278	312	...	...	...	127	154	180	206	233	272	...	129	147	166	185	213	241	70
75	399	116	152	188	224	261	297	333	...	...	...	136	164	192	220	248	290	...	137	157	177	197	227	257	75
80	413	124	163	201	240	279	318	...	...	...	...	145	175	205	235	265	...	...	147	169	190	212	244	276	80
85	425	131	172	213	254	295	...	...	...	...	...	153	184	216	247	279	...	...	156	179	201	224	258	292	85
90	438	139	182	225	269	312	...	...	...	...	...	162	195	228	261	295	...	...	165	189	213	237	273	309	90
95	449	146	191	236	282	327	...	...	...	...	...	170	205	240	275	...	...	...	173	198	223	248	286	323	95
100	461	153	201	248	295	...	...	...	...	...	...	179	215	252	288	...	...	...	182	208	235	261	300	...	100

63. 1⅜-IN. SMOOTH NOZZLE—2½- AND 3-IN. HOSE

Nozzle Pressure Indicated by Pitot Gage, lb. per sq. in.	Discharge, Gal. per Min.	Pressures Required at Hydrant or Fire Engine, while Stream is Flowing, to Maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality 2½- and 3-in. Rubber-Lined Hose																						Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
		Single 2½-in. Lines								Single 3-in. Lines							Two 2½-in. Lines Siamesed							
		100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	700 ft.	800 ft.	200 ft.	400 ft.	600 ft.	800 ft.	1,000 ft.	1,200 ft.	1,500 ft.	400 ft.	600 ft.	800 ft.	1,000 ft.	1,200 ft.	1,500 ft.	1,800 ft.	
20	250	37	52	68	83	98	113	128	144	34	45	57	68	80	92	109	37	46	54	63	71	84	96	20
25	280	46	64	83	102	121	139	158	177	41	56	70	85	99	113	135	46	57	67	78	88	104	119	25
30	307	55	77	99	121	144	166	188	210	50	67	84	101	118	135	161	56	68	81	93	106	124	143	30
35	331	64	89	115	140	166	191	217	242	58	78	97	117	137	157	187	65	80	94	108	122	143	165	35
40	354	73	102	131	160	189	218	247	276	67	89	112	134	157	180	214	74	90	106	122	138	162	186	40
45	376	81	114	146	178	211	243	275	307	74	99	125	150	175	200	238	83	101	119	137	155	182	209	45
50	396	90	125	161	196	222	257	293	328	82	109	137	164	192	220	267	92	111	131	151	171	201	230	50
55	415	99	137	176	215	254	292	331	....	90	121	151	182	212	242	288	100	122	144	165	187	219	252	55
60	434	107	149	191	233	276	318	....	....	98	131	163	196	229	262	....	109	133	156	180	203	238	273	60
65	451	116	161	206	251	297	....	....	....	106	141	177	212	247	282	....	118	143	168	194	219	257	294	65
70	469	125	173	222	270	319	....	....	....	114	152	189	227	265	303	....	128	155	182	209	236	277	317	70
75	485	134	185	237	289	....	....	....	....	122	162	203	243	283	....	....	137	165	194	223	252	295	....	75
80	500	142	196	251	305	....	....	....	....	130	172	215	257	300	....	....	145	175	206	236	266	312	....	80
85	516	151	209	267	325	....	....	....	....	138	183	229	274	....	....	....	153	186	218	250	282	331	....	85
90	531	159	220	281	....	....	....	....	....	146	194	241	289	....	....	....	162	196	230	264	298	....	....	90
95	546	168	232	297	....	....	....	....	....	153	203	254	304	....	....	....	170	206	241	277	313	....	....	95
100	560	177	244	312	....	....	....	....	....	162	215	267	....	....	....	....	179	217	254	291	329	....	....	100

64. 1½-IN. SMOOTH NOZZLE—2½- AND 3-IN. HOSE

Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.	Discharge, Gal. per Min.	Pressures Required at Hydrant or Fire Engine, while stream is flowing, to maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality 2½- and 3-in. Rubber-Lined Hose																							Nozzle Pressure Indicated by Pitot Gage. lb. per sq. in.
		Single 2½-in. Lines								Single 3-in. Lines							Two 2½-in. Lines Siamesed								
		100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	700 ft.	800 ft.	200 ft.	400 ft.	600 ft.	800 ft.	1,000 ft.	1,200 ft.	1,500 ft.	200 ft.	400 ft.	600 ft.	800 ft.	1,000 ft.	1,200 ft.	1,500 ft.	1,800 ft.	
20	298	44	65	86	107	128	149	170	191	39	55	71	87	104	120	144	33	45	56	68	79	91	108	126	20
25	333	54	80	106	132	158	184	210	236	48	68	88	108	128	148	178	41	56	70	84	99	113	135	156	25
30	365	65	95	126	157	188	219	250	280	58	81	105	129	153	177	212	49	66	83	100	117	134	160	185	30
35	394	75	110	145	181	216	251	287	322	67	94	122	149	177	204	245	57	77	96	116	135	155	184	214	35
40	422	85	126	166	206	246	286	327	....	76	107	139	170	201	232	279	65	88	110	132	155	177	211	244	40
45	447	96	141	185	230	275	320	....	....	85	120	155	189	224	258	....	73	97	122	146	171	196	233	269	45
50	472	106	155	205	254	304	....	....	....	95	133	171	209	247	286	....	81	108	136	163	190	218	259	300	50
55	494	116	170	224	278	332	....	....	....	104	145	187	228	270	....	....	88	118	148	178	208	237	282	327	55
60	517	126	184	242	301	....	....	....	....	113	158	203	248	293	....	....	96	128	161	193	225	257	305	....	60
65	537	136	198	261	324	....	....	....	....	122	170	218	267	....	....	....	104	139	174	208	243	278	....	....	65
70	558	146	213	281	....	....	....	....	....	131	183	235	287	....	....	....	112	149	186	223	261	298	....	....	70
75	578	156	228	299	....	....	....	....	....	140	196	251	307	....	....	....	120	160	199	239	279	319	....	....	75
80	596	166	242	318	....	....	....	....	....	149	208	267	....	....	....	....	127	170	212	254	296	....	....	....	80
85	614	176	257	337	....	....	....	....	....	158	220	282	....	....	....	....	135	179	224	268	313	....	....	....	85
90	633	187	272	....	....	....	....	....	....	167	233	298	....	....	....	....	143	190	237	284	....	....	....	....	90
95	650	197	286	....	....	....	....	....	....	176	245	314	....	....	....	....	152	201	251	301	....	....	....	....	95
100	667	207	300	....	....	....	....	....	....	185	257	....	....	....	....	....	160	212	264	316	....	....	....	....	100

65. 1⅝-IN. SMOOTH NOZZLE—2½- AND 3-IN. HOSE

Pressures Required at Hydrant or Fire Engine, while stream is flowing, to maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality 2½- and 3-in. Rubber-Lined Hose

Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.	Discharge, Gal. per Min.	Single 2½-in. Lines						Single 3-in. Lines						Two 2½-in. Lines Siamesed								Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
		100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	200 ft.	400 ft.	600 ft.	800 ft.	1,000 ft.	1,200 ft.	200 ft.	400 ft.	600 ft.	800 ft.	1,000 ft.	1,200 ft.	1,500 ft.	1,800 ft.	
20	350	52	80	108	136	165	193	46	68	90	112	134	156	37	53	68	84	100	115	139	162	20
25	392	65	100	135	170	205	240	57	84	111	128	165	192	47	66	85	104	123	143	171	200	25
30	429	77	118	160	201	242	284	68	100	132	164	196	228	56	79	102	125	148	171	205	240	30
35	463	89	136	184	231	279	326	78	115	152	189	226	263	65	91	117	144	170	197	236	276	35
40	496	101	155	208	262	316	....	89	131	173	215	257	299	74	104	134	164	194	224	269	314	40
45	525	113	173	233	293	....	....	100	146	193	239	286	....	82	116	149	182	215	248	298	....	45
50	554	125	192	258	324	....	....	111	162	214	265	....	....	91	128	165	202	239	275	331	....	50
55	581	137	210	282	....	....	....	121	178	234	290	....	....	100	140	181	221	261	301	....	....	55
60	607	149	228	306	....	....	....	132	193	254	....	....	....	109	153	196	240	283	327	....	....	60
65	631	162	246	330	....	....	....	143	209	275	....	....	....	118	164	211	258	305	....	....	....	65
70	655	173	263	....	....	....	....	153	223	294	....	....	....	126	176	226	276	326	....	....	....	70
75	678	184	281	....	....	....	....	163	237	312	....	....	....	135	189	242	295	....	....	....	....	75
80	700	197	299	....	....	....	....	174	253	....	....	....	....	144	201	258	314	....	....	....	....	80
85	722	209	317	....	....	....	....	184	269	....	....	....	....	153	213	273	....	....	....	....	....	85
90	743	220	....	....	....	....	....	195	284	....	....	....	....	162	225	289	....	....	....	....	....	90
95	763	232	....	....	....	....	....	205	299	....	....	....	....	170	237	303	....	....	....	....	....	95
100	783	244	....	....	....	....	....	216	314	....	....	....	....	179	249	319	....	....	....	....	....	100

66. 1¾-IN. SMOOTH NOZZLE—2½- AND 3-IN. HOSE

Pressures Required at Hydrant or Fire Engine, while Stream is Flowing, to Maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality 2½- and 3-in. Rubber-Lined Hose

Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.	Discharge, Gal. per Min.	Single 2½-in. Lines 100 ft.	200 ft.	300 ft.	400 ft.	Single 3-in. Lines 100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	800 ft.	1,000 ft.	Two 2½-in. Lines Siamesed 100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	800 ft.	1,000 ft.	1,200 ft.	Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
20	407	63	100	138	175	40	55	71	86	101	116	147	177	33	43	53	64	74	84	105	125	146	20
25	455	77	123	169	215	49	67	84	102	120	138	173	209	40	53	65	78	91	103	128	154	179	25
30	498	91	145	199	253	58	79	100	121	142	163	205	247	49	64	79	94	110	125	155	185	215	30
35	538	106	169	231	294	68	92	117	141	166	190	239	288	56	74	91	109	126	143	178	213	248	35
40	575	120	191	262	333	77	104	132	159	187	215	270	325	64	84	103	123	143	162	201	241	280	40
45	609	135	215	294	...	87	118	149	180	211	241	303	...	73	95	117	139	161	183	227	271	315	45
50	643	150	237	325	...	96	130	164	199	233	267	...	...	80	104	128	152	177	201	249	297	...	50
55	674	164	259	...	...	105	142	179	216	254	291	...	...	88	114	140	167	193	219	272	324	...	55
60	704	177	280	...	...	114	154	194	234	274	314	...	...	96	125	153	182	210	239	296	...	...	60
65	732	191	302	...	...	123	166	209	252	296	...	...	...	104	134	165	195	226	257	318	...	...	65
70	761	206	325	...	...	133	180	227	273	...	...	...	...	111	144	177	210	243	275	...	...	...	70
75	787	220	...	...	...	143	192	242	291	...	...	...	...	118	153	188	223	258	293	...	...	...	75
80	813	234	...	...	...	152	204	257	309	...	...	...	...	127	164	201	239	276	313	...	...	...	80
85	838	247	...	...	...	160	215	270	...	...	...	...	...	135	174	214	253	293	...	...	...	...	85
90	862	261	...	...	...	169	228	286	...	...	...	...	...	142	183	225	266	308	...	...	...	...	90
95	885	274	...	...	...	178	240	301	...	...	...	...	...	150	194	237	281	...	...	...	...	...	95
100	909	...	...	...	...	188	253	317	...	...	...	...	...	158	204	250	296	...	...	...	...	...	100

67. 2-IN. SMOOTH NOZZLE—2½- AND 3-IN. HOSE

Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.	Discharge, Gal. per Min.	Pressures Required at Hydrant or Fire Engine, while stream is flowing, to maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality 2½- and 3-in. Rubber-Lined Hose																			Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
		Single 2½-in. Lines			Single 3-in. Lines							Two 2½-in. Lines Siamesed								
		100 ft.	200 ft.	300 ft.	100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	800 ft.	100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	800 ft.	1,000 ft.	
20	532	90	152	214	52	76	100	124	148	172	220	41	58	75	92	110	127	161	195	20
25	594	111	187	263	65	94	123	152	182	211	270	51	72	93	114	135	156	198	240	25
30	651	132	222	312	77	112	147	181	216	251	321	61	86	110	135	160	185	234	284	30
35	703	152	255	....	89	129	169	209	249	289	....	71	100	128	157	186	214	271	329	35
40	752	173	290	....	102	147	193	238	283	....	....	81	113	146	178	211	243	308	....	40
45	797	193	323	....	113	163	213	263	314	....	....	90	126	162	198	234	270	....	....	45
50	841	214	....	....	126	182	237	293	....	....	....	100	140	180	220	260	300	....	....	50
55	881	....	....	....	138	199	260	321	....	....	....	110	153	197	240	284	....	....	....	55
60	920	....	....	....	150	216	282	....	....	....	....	119	166	213	260	308	....	....	....	60
65	958	....	....	....	162	233	304	....	....	....	....	129	180	230	281	....	....	....	....	65
70	994	....	....	....	175	251	327	....	....	....	....	139	193	248	302	....	....	....	....	70
75	1,029	....	....	....	187	268	....	....	....	....	....	148	206	264	322	....	....	....	....	75
80	1,063	....	....	....	199	285	....	....	....	....	....	158	219	280	....	....	....	....	....	80
85	1,095	....	....	....	211	302	....	....	....	....	....	167	232	297	....	....	....	....	....	85
90	1,128	....	....	....	223	319	....	....	....	....	....	177	245	314	....	....	....	....	....	90
95	1,158	....	....	....	235	335	....	....	....	....	....	186	258	....	....	....	....	....	....	95
100	1,189	....	....	....	247	....	....					196	272				....	....	....	100

68. 1¼-IN. SMOOTH NOZZLE—3½-IN. HOSE

Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.	Discharge, Gal. per Min.	Pressures Required at Hydrant or Fire Engine, while Stream is Flowing, to Maintain Nozzle Pressures Given in First Column, through Various Lengths of Best Quality 3½-in. Rubber-Lined Hose								Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
		600 ft.	700 ft.	800 ft.	900 ft.	1,000 ft.	1,200 ft.	1,500 ft.	1,800 ft.	
20	206	32	34	36	37	39	43	49	55	20
25	230	39	42	44	46	48	53	60	67	25
30	253	47	49	52	55	58	63	71	79	30
35	273	54	57	60	64	67	73	82	91	35
40	292	62	65	69	72	76	83	93	104	40
45	309	69	73	77	81	85	93	104	116	45
50	326	77	81	85	90	94	102	115	128	50
55	342	84	89	94	99	103	112	126	141	55
60	357	92	97	102	107	112	122	137	153	60
65	372	99	105	110	116	121	132	149	165	65
70	386	107	113	118	124	130	142	160	177	70
75	399	114	120	127	133	139	152	171	190	75
80	413	122	128	135	142	148	162	182	202	80
85	425	128	135	142	149	156	170	191	212	85
90	438	136	143	151	158	165	180	202	225	90
95	449	143	151	159	167	175	190	214	237	95
100	461	151	159	167	175	184	200	225	249	100

69. $1\frac{3}{8}$-IN. SMOOTH NOZZLE—$3\frac{1}{2}$-IN. HOSE

Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.	Discharge, Gal. per Min.	Pressures Required at Hydrant or Fire Engine, while stream is flowing, to maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality 3½-in. Rubber-Lined Hose									Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
		400 ft.	500 ft.	600 ft.	700 ft.	800 ft.	1,000 ft.	1,200 ft.	1,500 ft.	1,800 ft.	
20	250	31	34	36	39	41	47	52	60	67	20
25	280	39	42	45	49	52	59	65	75	85	25
30	307	46	50	54	58	62	70	78	89	101	30
35	331	54	58	63	67	72	81	90	103	117	35
40	354	61	66	71	76	81	91	101	116	131	40
45	376	69	74	80	85	91	102	113	130	147	45
50	396	76	82	88	95	101	113	126	144	163	50
55	415	84	90	97	104	111	124	138	158	179	55
60	434	91	98	106	113	121	135	150	172	195	60
65	451	98	106	114	122	130	146	161	185	209	65
70	469	106	114	123	131	140	157	174	199	225	70
75	485	113	122	131	140	149	167	185	212	239	75
80	500	120	130	140	149	159	178	197	226	255	80
85	516	127	138	148	158	168	188	208	239	269	85
90	531	135	146	156	167	178	199	221	253	285	90
95	546	142	153	165	176	187	209	232	266	299	95
100	560	150	161	173	185	197	220	244	279	315	100

70. 1½-IN. SMOOTH NOZZLE—3½-IN. HOSE

Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.	Discharge, Gal. per Min.	Pressures Required at Hydrant or Fire Engine, while stream is flowing, to maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality 3½-in. Rubber-Lined Hose								Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
		200 ft.	400 ft.	600 ft.	800 ft.	1,000 ft.	1,200 ft.	1,500 ft.	1,800 ft.	
20	298	28	36	43	50	58	65	76	87	20
25	333	35	44	53	62	71	80	93	107	25
30	365	42	53	63	74	85	96	112	128	30
35	394	49	61	73	86	98	111	129	148	35
40	422	55	69	83	97	111	125	146	167	40
45	447	62	78	93	109	125	140	164	187	45
50	472	69	86	103	121	138	155	181	207	50
55	494	76	94	113	132	151	170	198	226	55
60	517	82	102	123	143	163	183	214	244	60
65	537	89	111	133	154	176	198	231	263	65
70	558	96	119	143	166	189	213	248	283	70
75	578	103	128	153	178	203	228	265	303	75
80	596	109	136	162	188	215	241	281	......	80
85	614	116	144	172	200	228	256	298	......	85
90	633	123	152	182	211	241	271	.....	......	90
95	650	129	160	191	222	253	284	.....	......	95
100	667	136	168	201	233	265	298	.....	......	100

71. $1\frac{5}{8}$-IN. SMOOTH NOZZLE—$3\frac{1}{2}$-IN. HOSE

Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.	Discharge, Gal. per Min.	Pressures Required at Hydrant or Fire Engine, while stream is flowing, to Maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality $3\frac{1}{2}$-in. Rubber-Lined Hose								Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
		200 ft.	400 ft.	600 ft.	800 ft.	1,000 ft.	1,200 ft.	1,500 ft.	1,800 ft.	
20	350	31	41	50	60	70	80	94	109	20
25	392	38	51	63	75	87	99	118	136	25
30	429	46	60	75	89	103	118	139	161	30
35	463	53	70	86	103	120	136	161	186	35
40	496	61	79	98	117	136	155	183	211	40
45	525	68	89	110	131	152	173	205	236	45
50	554	76	99	122	145	168	192	226	261	50
55	581	83	108	133	158	184	209	247	284	55
60	607	90	117	144	172	199	226	267	308	60
65	631	97	127	156	186	215	244	289	......	65
70	655	105	136	167	199	230	262	309	......	70
75	678	112	145	179	212	245	279	.....	......	75
80	700	119	155	191	226	262	297	.....	......	80
85	722	127	165	202	240	278	316	.....	......	85
90	743	134	174	214	254	294	.....	.....	......	90
95	763	141	183	225	267	309	.....	.....	......	95
100	783	149	193	237	281	.....	.....	.....	......	100

72. 1¾-IN. SMOOTH NOZZLE—3½-IN. HOSE

Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.	Discharge, Gal. per Min.	Pressures Required at Hydrant or Fire Engine, while stream is flowing, to maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality 3½-in. Rubber-Lined Hose									Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
		100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	800 ft.	1,000 ft.	1,200 ft.	
20	407	28	35	41	48	54	61	74	87	101	20
25	455	35	43	51	59	67	75	91	107	123	25
30	498	41	51	60	70	79	89	108	127	146	30
35	538	48	59	70	81	92	103	124	146	168	35
40	575	55	67	80	92	105	117	142	167	191	40
45	609	62	75	89	103	117	131	158	186	213	45
50	643	68	84	99	115	130	145	176	206	237	50
55	674	75	92	109	125	142	159	192	225	259	55
60	704	82	100	118	136	154	172	208	244	280	60
65	732	89	108	127	147	166	186	224	263	302	65
70	761	95	116	137	158	178	199	241	282	.....	70
75	787	102	124	146	168	190	212	257	301	.....	75
80	813	109	132	156	179	203	226	273	320	.....	80
85	838	115	140	165	190	214	239	289	.....	.....	85
90	862	122	148	174	200	227	253	305	.....	.....	90
95	885	128	156	183	211	238	266	...	.....	.....	95
100	909	135	164	193	222	251	280	....	.....	.....	100

73. 2-IN. SMOOTH NOZZLE—3½-IN. HOSE

Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.	Discharge, Gal. per Min.	Pressures Required at Hydrant or Fire Engine, while stream is flowing, to Maintain Nozzle Pressures given in First Column, through various Lengths of Best Quality 3½-in. Rubber-Lined Hose								Nozzle Pressure Indicated by Pitot Gage lb. per sq. in.
		100 ft.	200 ft.	300 ft.	400 ft.	500 ft.	600 ft.	800 ft.	1,000 ft.	
20	532	33	44	55	65	76	87	109	130	20
25	594	41	54	67	80	93	106	133	159	25
30	651	49	64	80	96	111	127	158	189	30
35	703	57	75	93	111	129	147	183	219	35
40	752	65	85	105	126	146	166	207	247	40
45	797	72	95	118	140	163	185	231	276	45
50	841	80	105	130	155	180	205	255	305	50
55	881	88	116	143	170	197	225	279	......	55
60	920	96	126	155	185	214	244	303	......	60
65	958	104	136	168	200	232	263	.....	......	65
70	994	112	146	180	214	248	282	.....	......	70
75	1,029	119	156	192	229	265	301	.....	......	75
80	1,063	127	166	205	243	282	.....	.....	......	80
85	1,095	135	176	217	258	299	.....	.....	......	85
90	1,128	143	186	229	272	.....	.....	.....	......	90
95	1,158	151	196	241	286	.....	.....	.....	......	95
100	1,189	158	206	253	301	...	.....	.....	......	100

INDEX

INDEX

Note: Numbers in parenthesis ( ) refer to Sections; all other numbers refer to paragraphs.

For example: **(II)** 15, refers to Paragraph 15 of Section II. To find this reference turn to page marked Sec. **II**-15, at the outside top corner of the page.

Page numbers are not used in this index.

All paragraphs follow in numerical order but at some places skips occur in paragraph numbers.

E

F

N

O

R

S

T

U

V

LILLIBRIDGE—39-6071

Printed in U.S.A.

PRODUCTS OF WORTHINGTON

In addition to the complete pump line embraced by the contents of this book, Worthington also makes the products listed below:

COMPRESSORS

Air, Gas and Ammonia. Air Lifts

CONDENSERS

Surface, Jet, Barometric. Vacuum Auxiliaries. Steam-Air Ejectors

ENGINES

Oil and Gas

FEED-WATER HEATERS

Locomotive and Stationary

METERS

Water and Oil

Worthington products and patterns comprehensively cover the complete range for each line of manufacture, also their special service requirements. On the following pages the complete Worthington line of products is briefly described. There is literature covering all products and many of the uses to which they are put. Request for information pertaining to Worthington products and their applications will receive prompt attention.

Reciprocating Pumps: Steam pumps are made in simplex and duplex types; simple, compound- and triple-expansion, condensing and non-condensing, vertical and horizontal, in all sizes and designs. Among the more commonly used types are the following: General-service, turret and straight-way patterns, valve-plate and valve-pot types, piston and plunger types. Other types are also manufactured.

Power pumps for direct, geared or belt drive from electric motors, oil engines, etc., are manufactured in single-cylinder, duplex and triplex types, vertical and horizontal.

Pressure Reciprocating Pumps: Pot-valve-type pumps for high-pressure boiler feeding, hydraulic press and other services requiring high pressures are furnished for pressures up to 2000 lb. per sq. in. Forged liquid-end pumps are built for pressures up to 12,000 lb. per sq. in.

Centrifugal Pumps: Open-impeller pumps are made in sizes from 1 to 8 in. Single-stage and two-stage ball-bearing volute centrifugal pumps are made for all general-service requirements. These pumps are noted for their unusually high efficiencies—8 to 10 points higher than average centrifugal pumps designed for the same service.

Two- and four-stage volute pumps, for heads over 200 ft., have their impellers placed in opposed positions, which produces hydraulic balance and minimum end thrust. Sizes: 2 to 8 in.; capacities: 50 to 2500 gal. per min.

Multi-stage turbine pumps are built in all commonly used sizes and capacities.

Deep-Well Pumps: Worthington builds reciprocating and rotary types of deep-well pumps adapted to meet practically all deep-well pumping problems. Several of the types are of exclusive design and possess advantages not obtainable in any other type.

Feather (REG. U. S. PAT. OFF.) **Valve Compressors:** All Worthington air, gas and ammonia compressors are equipped with the FEATHER Valve, consisting of light strips of flexible ribbon steel which lift into recesses in the guard. The method of restraining the ends and the lightness of the strips preclude noisy, destructive impact found in most valves and insure long life, while their flexibility insures tightness of seating.

Horizontal compressors are available in the following types: Single-stage steam- or belt-driven, two-stage steam-driven or belt or direct-connected motor-driven, and two-stage uniflow steam-driven. The two-stage compressors are built in capacities up to 5000 cu. ft. per min. and are equipped with either three- or five-step variable capacity control.

Vertical belt-driven air compressors are made in air- and water-cooled types and are available in capacities up to 50 cu. ft. per min.

Full details are given in illustrated bulletins.

Condensing Equipment: Worthington can furnish steam-condensing apparatus to meet the requirements of any set of operating conditions that may exist. The line includes every type of jet condensing equipment, spirojector and barometric types as well as surface condensers, from the smallest sizes up to one hundred thousand-kilowatt turbine capacity, together with all vacuum auxiliaries, also steam-air ejectors.

Diesel Oil Engines: Worthington Diesel oil engines cover the entire range of Diesel-engine powers. The line includes vertical two-cycle, solid-injection type from 50 to 540 hp.; horizonal four-cycle air-injection type from 150 to 600 hp.; vertical four-cycle air-injection type from 300 to 1125 hp.; and vertical two-cycle, double-acting type—the latest oil-engine development—from 500 to 12,000 hp.

Meters: The Worthington line includes hot-water boiler-feed meters; meters of all types and sizes for handling cold water; and meters of the horizontal and vertical reciprocating positive displacement type for cold oil and for hot oil.

Feed-Water Heaters: The Worthington stationary feed-water heater is of the open type and is so constructed as to be easily maintained at maximum efficiency.

The Worthington locomotive feed-water heater, of the open type, represents the most effective means of feed-water heating yet devised for locomotives. Average results show that it removes 80 per cent corrosive oxygen from feed-water and that it saves 12 per cent in fuel and 14 per cent in total water per year.

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